THE

THE MARINE STEAM ENGI

R. SENNETT.
AND
H. J. ORA

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A TREATISE FOR ENGINEERING STUDENTS
YOUNG ENGINEERS, AND OFFICERS OF THE ROYAL NAVY
AND MERCANTILE MARINE

BY THE LATE

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AT THE ROYAL NAVAL COLLEGE; ETC.

WITH NUMEROUS DIAGRAMS

ELEVENTH EDITION

NEW IMPRESSION

LONGMANS, GREEN, AND CO.
39 PATERNOSTER ROW, LONDON
NEW YORK, BOMBAY, AND CALCUTTA
1918

Bibliographical Note

First Edition, March 1882.

Second Edition, with Additions and Corrections, October 1885.

Third Edition, Revised and largely Rewritten by H. J. Oram, January 1898.

Fourth to Tenth Editions, with Additions and Modifications, June 1899, *September* 1900, *April* 1902, *January* 1904, *October* 1906 *July* 1908, *and November* 1909.

Eleventh Edition, August 1911, *reprinted April* 1913.

PREFACE

TO

THE EIGHTH EDITION

THE first two editions of this work were written by the late Mr. Richard Sennett, but after his death, owing to great changes which had taken place in marine engineering, it required to be practically rewritten in order to modernise it. This task was undertaken by the writer in preference to preparing an independent new work, as there appeared to be advantageous features in its style and arrangement over others; but on account of revisions and additions made in several new editions to keep the book as far as possible up to date, little now remains of Mr. Sennett's work.

The printed matter and illustrations have been considerably increased from time to time as new editions were required, for it is generally agreed that ample illustration is essential to the proper understanding of written descriptions of engineering details.

As showing the considerable changes which have taken place since the writer's first association with the book, the chapters on water-tube boilers, oil-fuel arrangements, internal combustion engines, and the marine steam-turbine may be mentioned.

The importance of water-tube boilers has caused this subject to be dealt with at considerable length, while the chapter on the marine steam-turbine has in this edition been largely amplified

and the details of the Parsons type fully illustrated to keep pace with this departure in marine engineering, the rapidly increasing importance of which will be realised when it is stated that all new ships for the British Navy from battleships to torpedo-boat destroyers are being fitted with the Parsons' marine steam-turbine. As regards this chapter I acknowledge with many thanks assistance which has been given me by the Hon. C. A. Parsons and his staff.

The chapter on internal combustion engines is inserted in this edition for the first time, as it appears desirable that a short account of oil and gas engines should be given owing to their advent for various purposes on ships and boats and their possibilities as regards the future.

The care and management of marine engines and boilers have been dealt with in considerable detail, and it is hoped this will be of value to young engineers, although the duties of engineers in this respect cannot be adequately learnt from books, for actual experience in the engine rooms will alone completely supply the requisite instruction. What is given will, however, prepare young engineers for such experience, and give information on points which they are at first more or less unacquainted with.

In the preparation of a work of this kind, one becomes indebted in various ways to many friends, and to these the writer tenders his thanks, especially to Engineer Commander P. Marrack, R.N., Engineer Inspector at the Admiralty.

LONDON, 1906. H. J. ORAM.

NOTE TO ELEVENTH EDITION.

VARIOUS additions have been made in this edition, especially in those chapters dealing with the Steam Turbine, the Torsion Meter, and the Internal Combustion Engine, as required by their development and increasing importance in Marine Engineering.

July, 1911.

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MARINE STEAM-ENGINE.

CHAPTER I.

HISTORY AND PROGRESS.

The earliest steam-engines were simply reciprocating engines, and for many purposes such engines are still used even at the present day. Until, however, a suitable method of turning reciprocating into rotative motion had been discovered and utilised not any progress was made in adapting the steam-engine to the propulsion of vessels. The adoption of the crank effected this desirable object, enabled the power of the engine to be transmitted to the propeller smoothly and without shock, and was an indispensable step in the progress of steam navigation.

The *marine* steam-engine may justly be considered as a production of the nineteenth century. In the latter part of the eighteenth century several attempts were made to adapt the steam-engine for the propulsion of boats, but none of them were quite successful. The first practical steamboat was built on the Clyde, in 1801, by William Symington, for Lord Dundas. She was called the 'Charlotte Dundas,' and was worked for some time with success as a tug on the Forth and Clyde Canal, but was withdrawn from this service in consequence of an apprehension that the banks of the canal would suffer from the wash of the propeller. This boat was fitted with a single paddle-wheel placed near the stern, driven by a horizontal direct-acting engine, with connecting-rod and crank, and the general arrangement of her machinery would be considered creditable even at the present day.

The first recorded instance of steam navigation proving commercially successful was in America, where, in 1807, Robert Fulton built a steam vessel called the 'Clermont,' propelled by paddles driven by a Boulton & Watt engine. In 1812 Henry Bell built a vessel called the 'Comet,' which was successfully worked on the Clyde as a passenger steamer between Glasgow and Greenock. The 'Comet' was propelled by two pairs of paddles, each paddle having four floats or blades, somewhat resembling a pair of canoe paddles, crossed at right angles. The paddles were driven by an engine of somewhat peculiar design, which, however, approximated to the side-lever engine of a later day. This small boat was the first passenger steamer in Europe.

From this date the success of steam navigation may be said to have been secured, and the advancement that has been made since has not consisted so much in the discovery of new principles as in the extension of old ones, and the introduction and development of improved mechanism and workmanship, with consequent economy of fuel. The result has been a progressive increase in the size, power, and speed of steamships and in the extent of their voyages; this increase culminating in the new fast steamships of the Cunard Steamship Company for the Atlantic voyage, which will displace over 35,000 tons, and be capable of being driven at a speed of 25 knots by engines of over 60,000 indicated horse-power.

Side-lever engine.—The propeller used in the earlier steamships was invariably the paddle-wheel, and the type of engine existing and giving satisfaction on land was naturally adapted at first to rotate

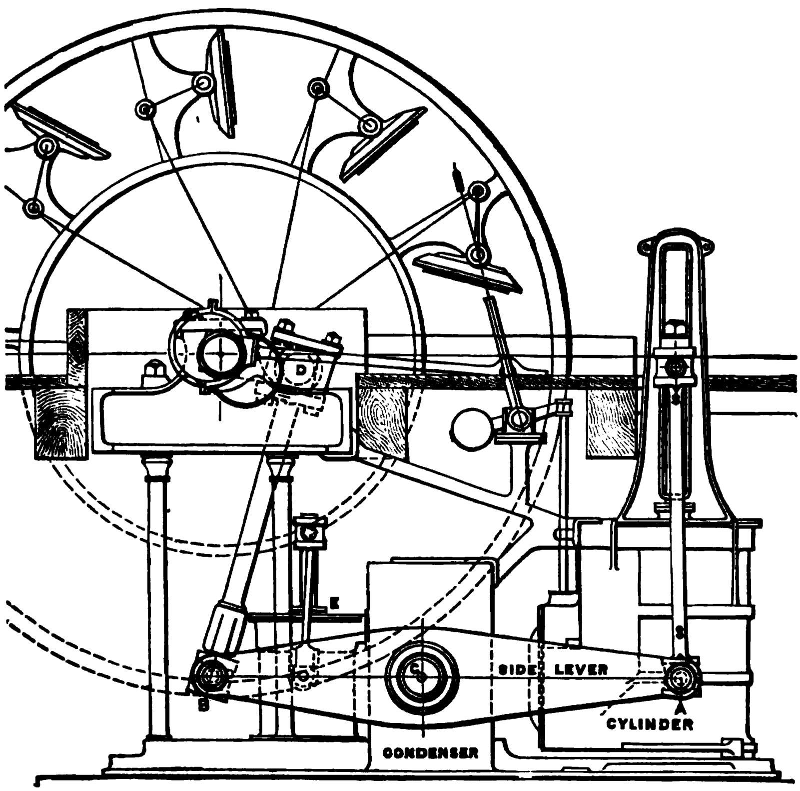


FIG. 1.

these paddle-wheels. Almost all these early engines were, therefore, of the *beam* type. In America the beam was generally placed over the crank, while in this country it was placed below the crank. The latter type of engine was known as the *side-lever engine*. The

general arrangement of the side-lever engine is shown in outline in Fig. 1, and it represents the type usually fitted not only in the first steam vessels, but also for some years after.

On the top of the piston-rod is fixed a crosshead with side-rods, s, attached at each end, which, passing down on either side of the cylinder, are connected to the ends, A, of a beam or side-lever, A B, oscillating on a fulcrum or gudgeon at its centre, C. The opposite ends, B, of these side-levers are fitted with journals carrying the crosstail, to the centre of which, one end of the connecting-rod B D is attached, the other end working on the crankpin D. The air-pump E is worked by side-rods from intermediate points in the side-levers, the upper ends of the air-pump side-rods being jointed to the opposite ends of the air-pump crosshead, to the centre of which the air-pump rod is secured. The piston-rod crosshead works in vertical guides to insure parallelism, and the parallel-motion rods used in land beam engines are dispensed with.

Grasshopper engines.—The arrangement of the side-levers was sometimes varied by making them levers of the third order, the gudgeon or fulcrum being at one end and the steam cylinder placed between the gudgeon and connecting-rod. These engines were commonly known as *grasshopper engines*.

The side-lever type of engine, though very heavy and occupying a large space for the power developed, was safe and reliable, securing a sufficient length of connecting-rod, and having its moving parts practically in equilibrium. It consequently continued in general use for a great number of years, but was at length superseded by the direct-acting type, which was lighter and more compact.

Introduction of steam war vessels.—Steam vessels were introduced into the Royal Navy in the year 1820, when the 'Monkey,' a vessel of 210 tons, was built at Rotherhithe and fitted by Messrs. Boulton & Watt with engines of 80 nominal horse-power. There were two cylinders, about 35½ in. diameter and 3 ft. 6 in. stroke, working at 26½ revolutions per minute, giving a mean piston speed of 185 ft. per minute. She was followed in 1822 by the 'Active,' of 80 nominal horse-power, by the same firm, and in 1823 by the 'Lightning,' of 100 horse-power, by Messrs. Maudslay, and some others whose names appeared for the first time in the Official Navy List for March 1828. These early steam vessels were mainly used for towing and general purposes, and could scarcely be classed as war vessels. Between this date and 1840 seventy other steam vessels were added to the Navy, the majority being fitted with flue boilers and slow-moving side-lever engines worked with steam at a pressure of 4 lbs. per square inch above the atmosphere.

The 'Rhadamanthus,' one of these ships, was fitted with side-lever engines and flue boilers by Messrs. Maudslay, Sons, & Field in 1832. The nominal horse-power was 220, but the engines were capable of being worked up to 400 I.H.P., or 1·8 times the nominal power. The load on the safety valves was 4 lbs. per square inch, and the number of revolutions per minute when working at full power 17½, giving a mean piston speed of 175 ft. per minute. The total weight of the machinery was 275 tons, or 13·75 cwts. per I.H.P. developed.

Between 1840 and 1850 tubular boilers were introduced. In these

boilers a group of small tubes was substituted for the long winding flue, to convey the heated gases from the furnaces to the chimney. The boilers were thus made lighter and more compact, and the working pressures of steam generally were increased to from 10 to 15 lbs. per square inch above the atmosphere.

Abandonment of side-lever engines.—Attempts were soon made to reduce the space required by the machinery, and the side-levers were abandoned and *direct-acting engines* fitted for rotating the paddle-wheels. Several arrangements of this kind were fitted, the two best known being the double-cylinder engine by Messrs. Maudslay and the oscillating engine adopted by Messrs. Penn. Fig. 2 shows the *double-cylinder engine*, which consisted of two equal cylinders side by side, the piston-rods from the two cylinders being connected to a single cross-head. In order to get sufficient length of connecting-rod, the cross-head was of peculiar form and passed down between the cylinders, having a journal at its lower end, on which one end of the connecting-rod worked, the other end being attached to the crankpin.

Fig. 3 shows the general arrangement of the *oscillating engine*, which is the simplest and most compact type for driving paddle-wheels. This type of engine, although first fitted for marine purposes by Messrs. Maudslay, Sons, & Field, who in 1828 fitted a pair of oscillating engines into the steamship 'Endeavour,' and subsequently in several other ships, was adopted and perfected by the late eminent engineer, Mr. John Penn, with whose name it is now generally associated. In these engines the connecting-rod is altogether dispensed with, the upper end of the piston-rod being fitted with brasses to work directly on the crankpin, and the cylinder itself is carried on trunnion bearings, to allow the necessary oscillation to suit the motion of the crank. The trunnions are hollow, and the steam is admitted to and exhausted from the cylinders through them. In this type of engine space and weight have been economised as far as is possible for paddle-wheel engines, and the majority of engines now made for paddle-wheel vessels are on this plan.

The 'Magicienne' was one of the best specimens of the steam war-vessels of that period. She was fitted with oscillating engines by Messrs. Penn in 1850. The pressure of steam in the boilers was 14 lbs. per square inch, number of revolutions per minute at full power 20½, giving a mean piston speed of 287 ft. per minute, with a maximum I.H.P. of 1,300. The total weight of the machinery was 275 tons, or 4·23 cwts. per I.H.P.

Defects of paddle-wheels.—The paddle-wheel possessed many practical disadvantages which interfered with progress beyond a certain point. Its performance was much affected by the variation of draught of the ship during a voyage, as the coal and stores were consumed, and the paddle-boxes offered resistance to the progress of the vessel. For fighting ships paddle-wheels were particularly unsuitable. The wheels themselves were exposed to danger from shot and shell, and the paddle-boxes interfered seriously with the training and working of the guns, while the shafting and many parts of the engines had to be considerably above the water-line, much of it above the upper deck. The paddle-wheel also is not a form of propeller well adapted for the application of high powers.

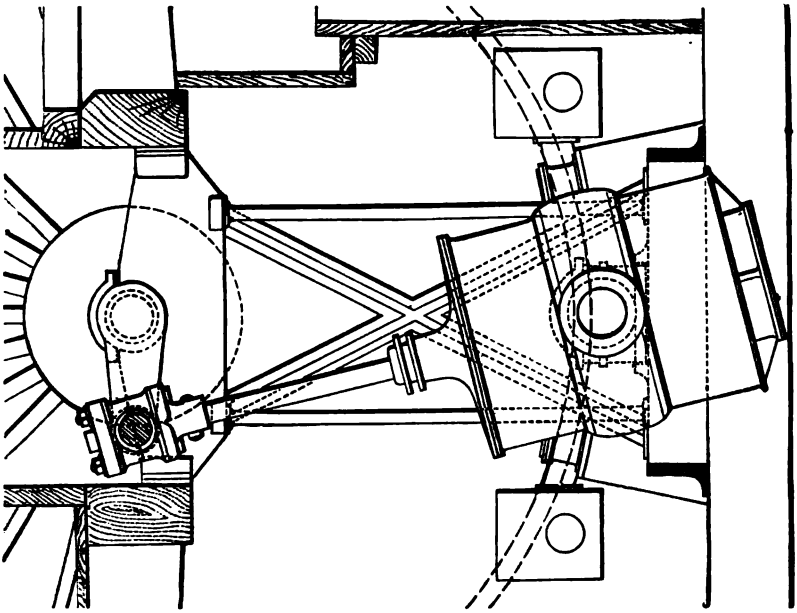


FIG. 3.

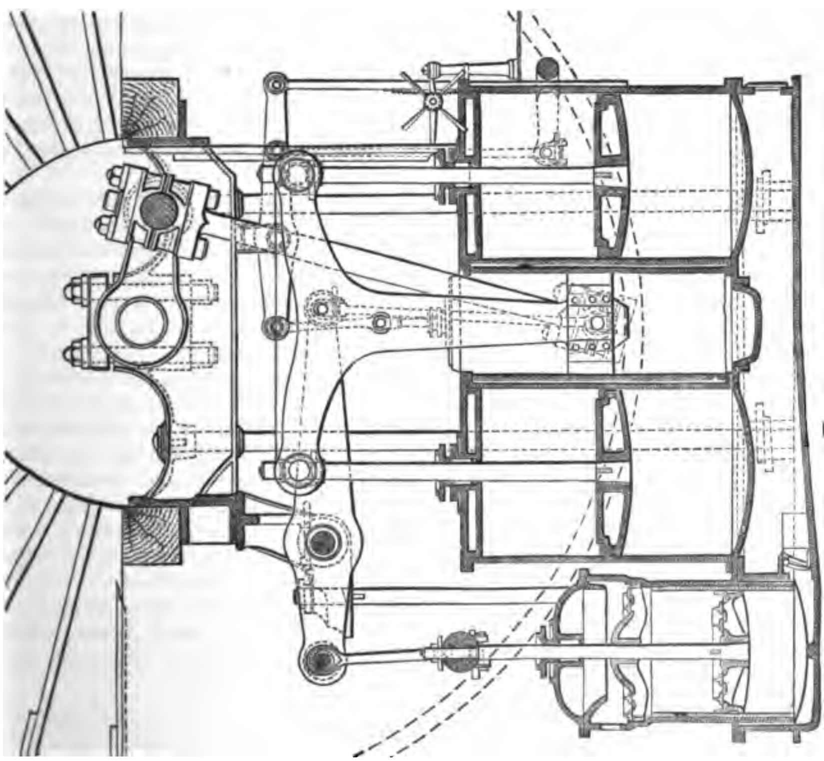


FIG. 2.

Adoption of the screw propeller.—The adoption of the screw propeller in lieu of the paddle-wheel was the most important step taken in the progress of marine engineering, for this rendered many subsequent advances possible. Its principal advantages, as compared with the paddle-wheel, are, that it is comparatively little affected by the rolling, or by the variation of the draught of the ship during a voyage, and it is equally capable of application to either great or small powers. It is not exposed to damage by projectiles, and also permits of the engines being kept below the water-line, which is very important in unarmoured warships. Screw engines, whether horizontal or vertical, can be further protected, if necessary, by being kept below the steel armour deck, with armour gratings in the necessary engine-room hatchways and other openings, while in the larger class of war vessels, such as the battleships and large cruisers, their height is so moderate that efficient protection can be given them by armour, even when the engines are vertical and the cylinders above the water-line. With screw engines the decks are also kept clear for the guns.

The substitution of the screw propeller for the paddle-wheel began to grow general about the period 1845–50. The screw propeller had been invented long before, but its practical utility had not been generally recognised, and it was still regarded as being in the experimental stage. The first notable experiments as to the comparative efficiencies of paddle-wheels and screw propellers were made in 1840, when the 'Archimedes,' with a screw propeller, beat the paddle-wheel boat 'Ariel' between Dover and Calais by five to six minutes under steam and sail. The 'Archimedes' afterwards beat the paddle-wheel steamers 'Beaver' and 'Swallow,' but was beaten slightly by the 'Widgeon.' The Admiralty, in 1843, caused some important experiments to be carried out with the screw ship 'Rattler' and the paddle-wheel ship 'Alecto,' and, in 1849, with the screw ship 'Niger' and paddle-wheel vessel 'Basilisk.' The results in each case were in favour of the screw propeller, and many valuable conclusions were deduced from the trials.

From that time the use of the screw propeller gradually became more general, till at the present day it is almost solely employed for marine propulsion, the paddle-wheel only being applied in special cases. It is not too much to say that ships of the class now traversing the ocean in all directions, both in the royal and mercantile navies, would not have been possible had not the screw superseded the paddle.

Gearing for screw engines and its abandonment.—In order to attain the same speed of ship the screw propeller had to be driven at a much greater speed than the paddle-wheel, and as it was not possible in the then condition of mechanical engineering to drive the pistons at a sufficiently high speed to enable the engine shaft to be connected directly to the propeller shafting, the earlier engines used for working screw propellers were *geared*, so that the screw shaft was caused to revolve at a much higher rate of speed than the engine shaft. A large spur wheel, keyed on the crank-shaft of the engine, worked into a pinion on the screw propeller shafting, so that the speed of the engine shaft could be multiplied on the screw shaft as might be necessary.

Before long, however, such improvements in workmanship and mechanical details were effected, that the speeds both of piston and

of revolution could be sufficiently increased to allow *direct engines* to be fitted. In these the gearing is left out, and the crank-shaft connected direct to the screw shafting. In many marine engines at the present day, even of the largest size, the mean piston speeds are as high as from 800 to 950 ft. per minute at the maximum power, while for torpedo boats and destroyers it rises as high as 1,200 ft. per minute, and in extreme cases to 1,400 ft. With the introduction of steam turbines gearing has in some ships again been introduced. In these cases the gearing is fitted to enable the speed of propeller to be reduced as compared with the high speed of rotation of the turbine.

Horizontal engines.—The paddle-wheel engines were either vertical or inclined; but when the screw propeller was introduced, and it became possible to place the whole of the propelling apparatus below the water-line, the engine was placed horizontally, and from that time, for about thirty years, the engines of warships were almost always of the horizontal type. One of the great obstacles that had then to be overcome in connecting the crank-shaft of the horizontal engine direct to the screw shafting was the close proximity in which the cylinder was necessarily placed to the centre-line of the ship, owing to the limitation of the beam of the ship, which made it difficult to get a connecting-rod of suitable length to work between the cylinder and the crank.

Trunk engines.—Mr. John Penn solved this difficulty by his invention of the trunk engine. In this engine a large hollow trunk, cast on or bolted to the piston, and working through a steam-tight stuffing-box on the end of the cylinder, was substituted for the piston-rod, and the connecting-rod was attached directly to a journal or gudgeon in the centre of the piston itself, as shown in Fig. 4. Though the use of a large trunk of this description does not at first sight appear desirable, yet the engines of this type have generally worked in a satisfactory manner, and they were amongst the most smooth-working and efficient marine engines employed. With the introduction of high-pressure steam, however, they became obsolete, owing to the difficulty of keeping the trunks in a steam-tight condition.

Return connecting-rod engines.—This kind of engine was adopted by the majority of marine engineering firms to enable the horizontal cylinders to be brought close to the crank-shaft, and, as usually fitted, is shown in Fig. 5. There were two rods to each piston, one passing above, the other below the crank-shaft, to the opposite side of the ship, while the further ends of the piston-rods were fixed to a cross-head, having a journal at its centre, from which the connecting-rod worked back to the crank.

In some later examples, in order to obviate the disadvantage of having more than one stuffing-box for each cylinder, and simplify the design of the piston, a single piston-rod was fitted, attached to a cross-head between the cylinder and the crank-shaft, from which two rods were carried, one above, the other below, the shaft, to a similar cross-head on the opposite side, as in the ordinary return connecting-rod arrangement.

Direct-acting engines.—The *direct-acting engine* shown in Fig. 6, having the connecting-rod between the cylinder and the crank, was often employed, especially by Messrs. Humphrys, in the later horizontal examples, the parts being stowed as compactly as possible in the limited

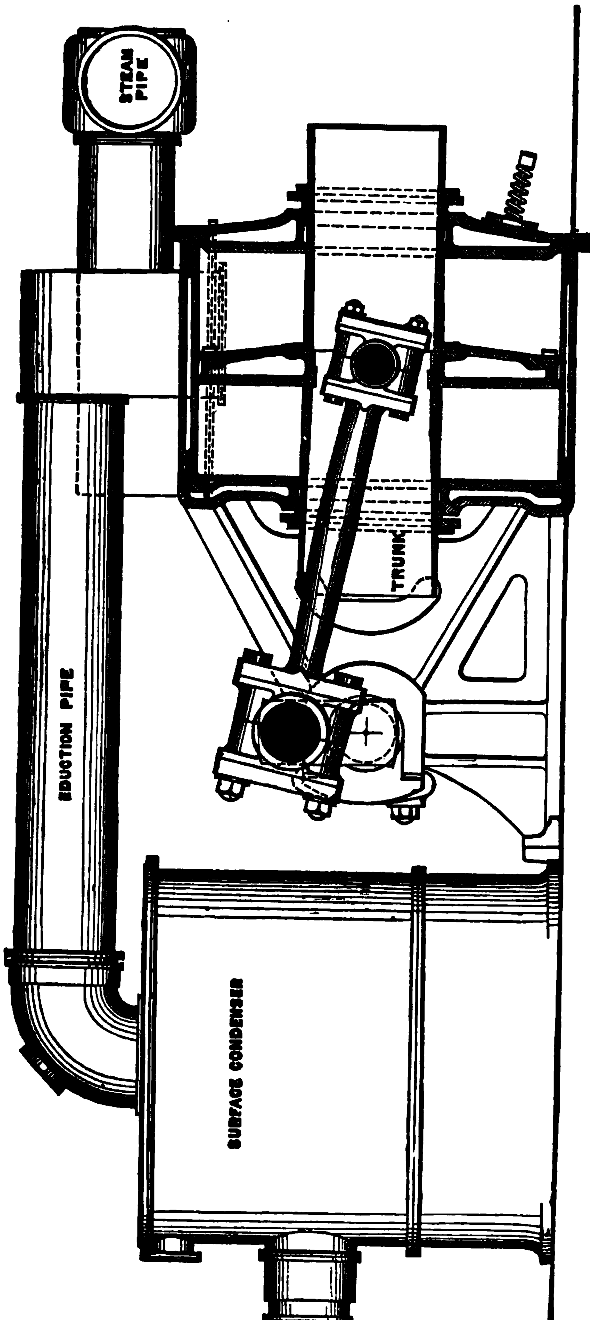


Fig. 4.

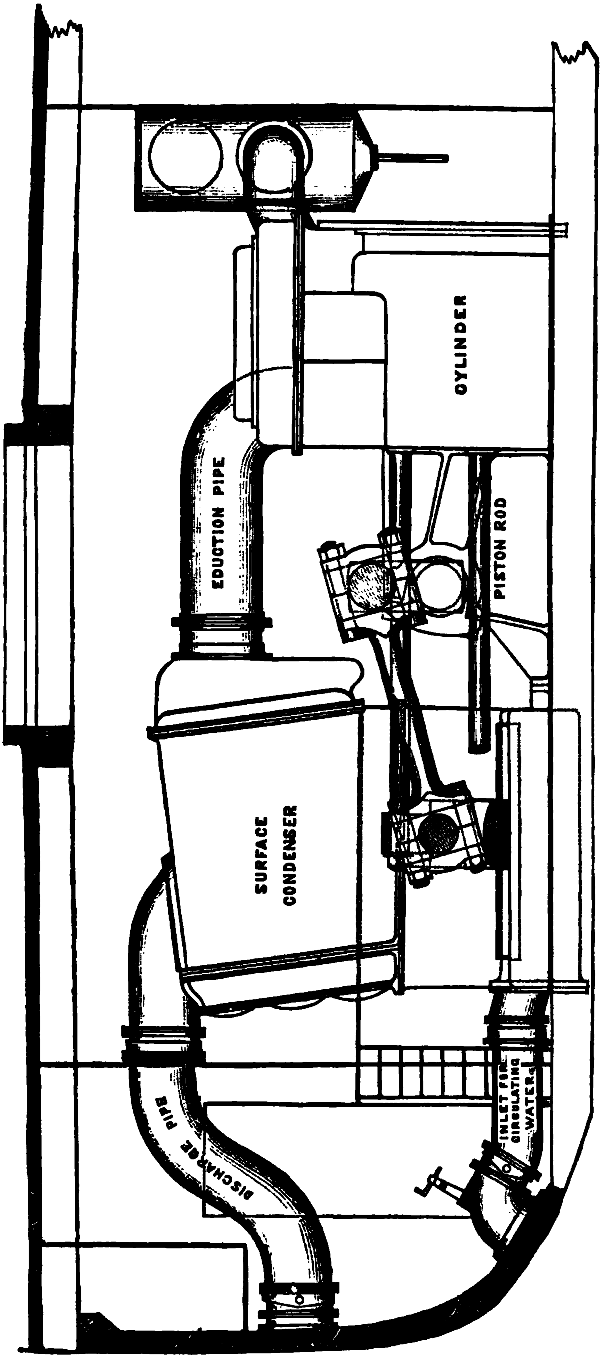

CYLINDER
EDUCTION PIPE
PISTON ROD
SURFACE
CONDENSER
DISCHARGE PIPE
INLET FOR
CIRCULATING
WATER

Fig. 5.

space available, and a short connecting-rod fitted. It is the simplest type, and the most suitable for general work, and, whenever sufficient room can be obtained, it is usually adopted. For vertical engines, with the cylinders at the top working down to the crank-shaft, which are now generally fitted for marine purposes, this type is universally adopted.

Early screw engines.—The majority of steamers, both war and mercantile, built during the years 1850–60, were fitted with horizontal screw propeller engines worked with steam of from 20 to 25 lbs. pressure per square inch. The engines had jet injection condensers, and were not remarkable for economy of fuel, but they were much lighter, and occupied considerably less space, than the paddle-wheel engines that preceded them. The mean piston speed in this type of engine was generally about 400 ft. per minute, and the weight of machinery about 3¾ cwts. per I.H.P.

Surface condensation.—The adoption of surface condensation, which became general about 1860, formed a most important step in marine engineering. Its value consisted not so much in the economy effected by the avoidance of loss from the brining of boilers, as in the fact that by its eliminating the element of danger resulting from deposit of solid non-conducting matter on the heating surfaces, it rendered possible the use of higher steam pressures in marine boilers, and led eventually to the introduction of cylindrical boilers and compound engines. When surface condensation was first introduced, the old flat-sided boilers, made to fit the section of the ship, were still retained, but were strengthened by fitting additional stays to enable them to carry steam pressures of 30 to 35 lbs. per square inch, and the majority of warships built during the years 1860–70 were fitted with surface-condensing engines worked with steam of this pressure. The piston speeds were also considerably increased, especially in the larger ships in which a long stroke could be obtained. With this type of engine the mean piston speeds varied from 500 to 665 ft. per minute. To promote economy of fuel the cylinders were usually steam-jacketed, and made large enough to allow for considerable expansion at full power, and the boilers were fitted with superheaters. The average weight of the machinery of this type, including the water in boilers and condensers, was about 3 cwts. per I.H.P.

Compound or double expansion engines.—After the introduction of the surface condenser, attention was directed to the use of higher steam pressures and greater expansion of steam, as theoretical considerations showed that considerable gain could thus be effected. The result was that the steam pressure was increased from 30 or 35 lbs. to 60 lbs., while cylindrical boilers were fitted to safely carry the increased pressure, and the engine was changed to the compound type. Compound engines were fitted to nearly all warships from 1870 to 1885.

In this type of engine the expansion is conducted in stages; the steam, after being admitted to a small cylinder and expanding therein, is led to a larger cylinder, where it expands still further prior to exhaust, so that the stresses on the framing and journals are decreased and the loss from liquefaction of steam in the cylinders reduced

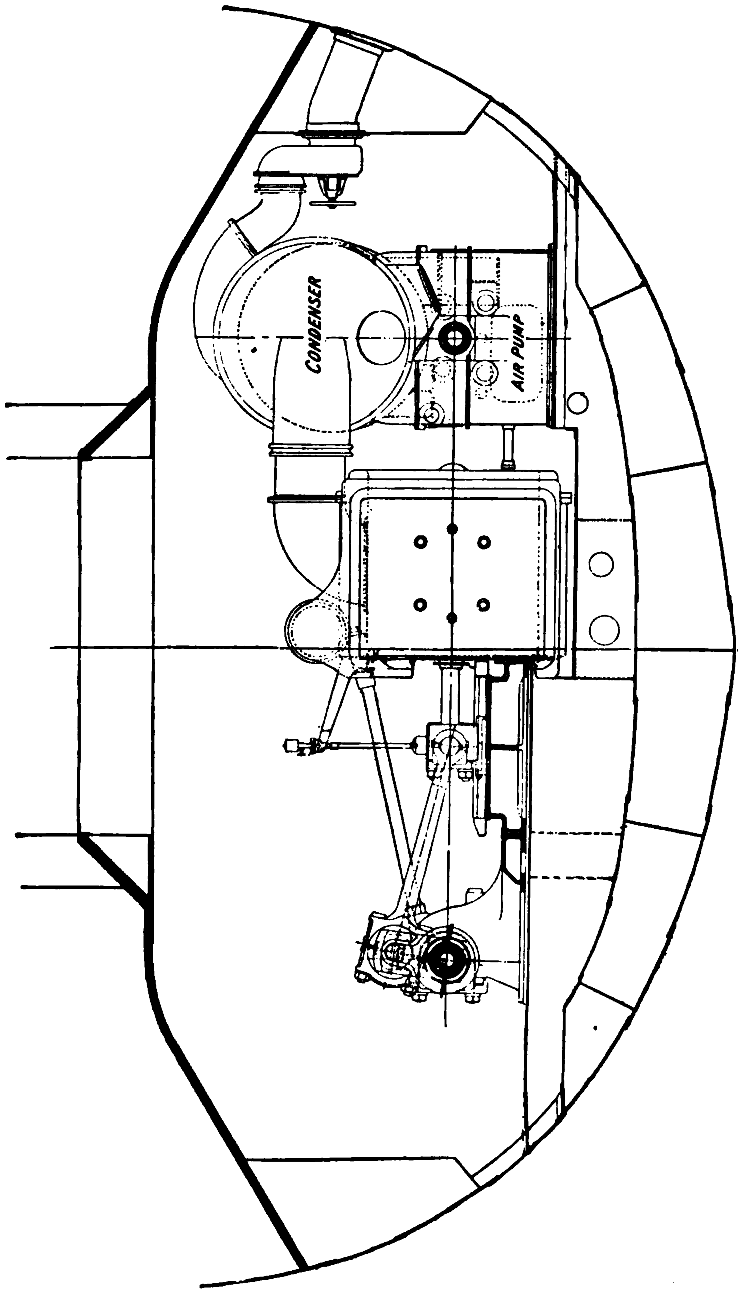


FIG. 6.

to a minimum. The following summary of its advantages is confirmed by experience :—

1. Reduction of the maximum stresses on the framing, shafting, and bearings, and consequent reduction of weight and cost.

2. Increased regularity of turning moment, and consequent increased efficiency of the propeller in the water.

3. More economical use of the steam in the cylinders and consequent increase of power from a given expenditure of heat.

The working steam pressure in the Royal Navy with this type of engine was originally 60 lbs. per square inch. This has been gradually increased from time to time, till in about the year 1880 it was 90 lbs., while in the last of this type fitted the pressure was increased to 120 lbs.

From the adoption of compound engines and higher steam pressures a considerable economy of fuel at once resulted. The gain in economy by the use even of the 60-lb. compound engines over the ordinary surface-condensing engines worked with steam of 30 lbs. pressure may be taken to be at least 30 per cent.

This gain was well authenticated, and the average amounts claimed by the principal Engineers and Steamship Companies, in reply to questions by an Admiralty Committee in 1872, was 30 to 35 per cent.

Vertical engines.—The vertical type of engine, with cylinders at the top and crank-shaft below, was adopted for merchant ships long before it was introduced into the Royal Navy, because it was a necessity in most warships that all the machinery should be kept below the water-line, and horizontal engines alone satisfied this condition. Figs. 7 and 8 show a vertical engine of the type fitted in the mercantile marine. Vertical engines possess many practical advantages over horizontal engines, especially in connection with the working of the cylinders and pistons, and general accessibility of the engine. When, therefore, the twin-screw system was adopted for armour-clad ships, vertical compound engines were fitted, with a middle line water-tight bulkhead separating the two sets. By dividing the power into two parts, each set of engines, even in a ship of great power, would be of moderate dimensions, and although the whole of the machinery might not in all cases be entirely below the water-line, the parts above would be protected, not only by armour plating, but by a body of coal in addition, the coal-bunkers being continued on each side of the engine room. This extension of the use of vertical engines has continued and been applied to all classes of vessel, and special means for protecting the cylinders have often been fitted. All new engines for the Navy, prior to the adoption of the steam turbine, were made vertical for all classes of vessel.

Three-cylinder compound engines.—As the power of compound engines increased the dimensions of the low-pressure cylinders became so great that it was found desirable to fit two low-pressure cylinders instead of one, in consequence of the difficulties experienced in obtaining sound castings of large size, and to keep the size of the reciprocating parts as small as possible. This led to what is known as the *three-cylinder compound engine*, which is simply a modification of the ordinary two-cylinder compound engine. Figs. 9 and 10 show a vertical compound engine of the three-cylinder type.

Triple expansion engines.—With initial steam pressures above

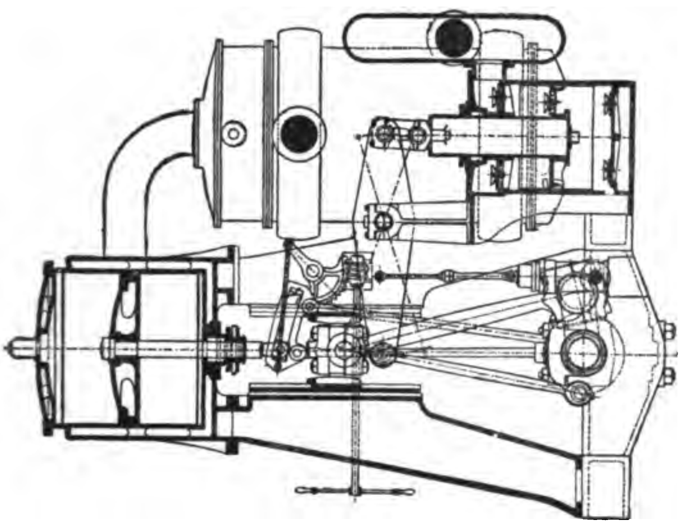


Fig. 8.

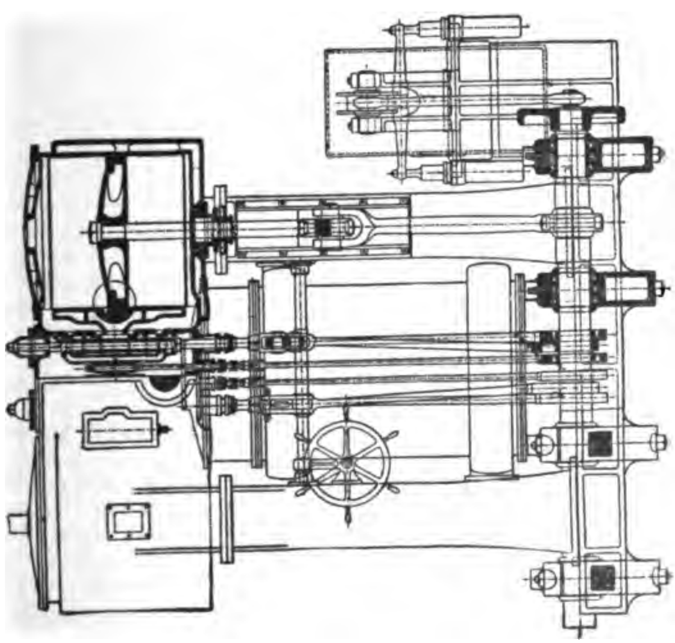


Fig. 7.

100 lbs. per square inch, the variation of temperature in each cylinder of an ordinary compound engine again becomes great, so that the full economy due to the high pressure cannot be attained in consequence of the loss from liquefaction. It was therefore soon found desirable to extend the compound system, and divide the expansion into three stages, carried out in separate cylinders, so as to reduce the range of temperature in each.

Engines on this system are usually known as *triple expansion* or *triple compound engines.* They were first introduced by the late Dr., then Mr., A. C. Kirk, of Messrs. R. Napier & Sons, Glasgow, who, in 1874, fitted them on board the s.s. 'Propontis,' to utilise steam of 150 lbs. pressure, supplied by Rowan & Horton's water-tube boilers. These engines gave good economical results, but the boilers unfortunately gave trouble, and were ultimately taken out. Very little further was done in this direction, until, in 1881, Mr. Kirk fitted a set of triple expansion engines on board the s.s. 'Aberdeen,' for the trade to Australia and China. The results in this instance were so satisfactory that other engines of the same type followed, and the system was soon largely adopted in the mercantile marine. Since 1885 the new ships for the Royal Navy have been fitted with triple expansion engines, which type is now the most general for marine purposes. The steam pressure first used with them in the Navy was 130 lbs., which was gradually increased to 155 lbs. in the year 1887. From this date to 1895 large numbers of triple expansion engines were added to the Navy, all with 155 lbs. steam pressure.

In the two large cruisers 'Powerful' and 'Terrible,' commenced in 1893, and tried in 1896-97, a boiler pressure of 260 lbs. was adopted, reduced to 210 lbs. at the engines, while in cruisers with reciprocating engines, of 1895, and subsequently, these pressures have been increased to 300 and 250 lbs. respectively. Triple expansion engines were fitted, the low-pressure cylinders being divided into two parts.

The gain in economy effected by the triple expansion engine, worked with steam of 130 lbs. to 150 lbs. pressure, over the ordinary compound engine worked at 90 to 100 lbs. pressure, may be taken at from 15 to 20 per cent., while with the higher pressures it will be still greater. Figs. 11 and 12 show the general arrangement of a triple expansion engine. In the mercantile marine, 180 lbs. steam pressure is the average now used in new ships, 210 lbs. being sometimes supplied.

Quadruple expansion engines.—In many cases in the mercantile marine the stage expansion principle is carried still further, and quadruple expansion engines fitted, dividing the expansion into four stages, the boiler pressures averaging in new ships 215 lbs. per square inch, and being 267 lbs. per square inch in one example.

These engines are more suitable for the mercantile marine, where the range of powers required from the engines is limited, than for the Navy, where this range is large; also as regards the Navy generally, evidence does not show that the additional complication thus introduced, and the extra length of engine room required, together with the additional engine friction, is compensated for by a sufficient gain in economy. They are gradually being introduced into the mercantile marine, but in the Navy only one torpedo boat and some smaller craft have been so fitted.

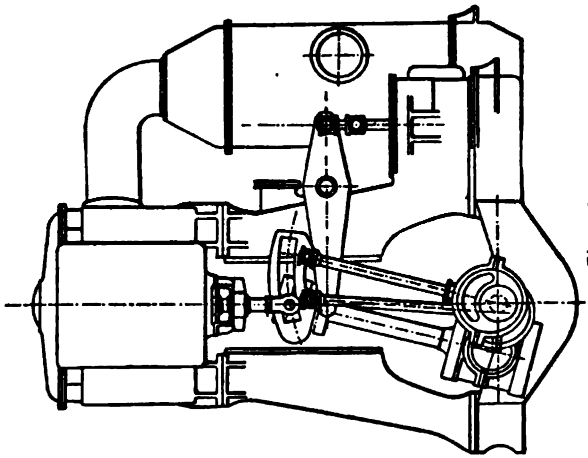


Fig. 10.

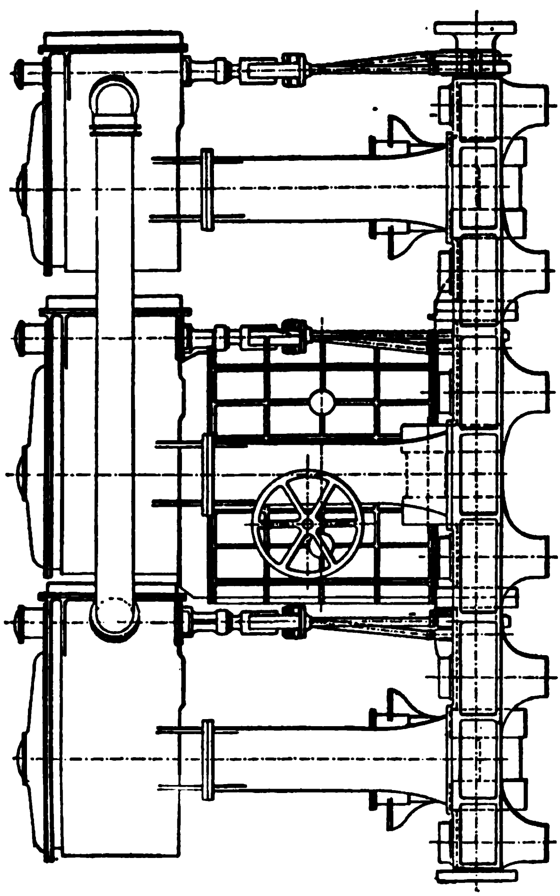


Fig 9.

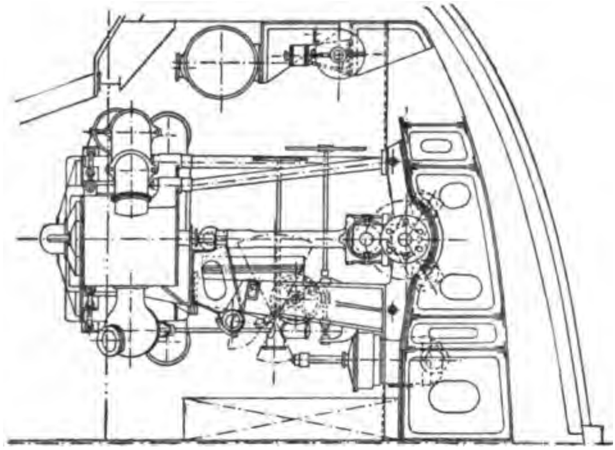


FIG. 12.

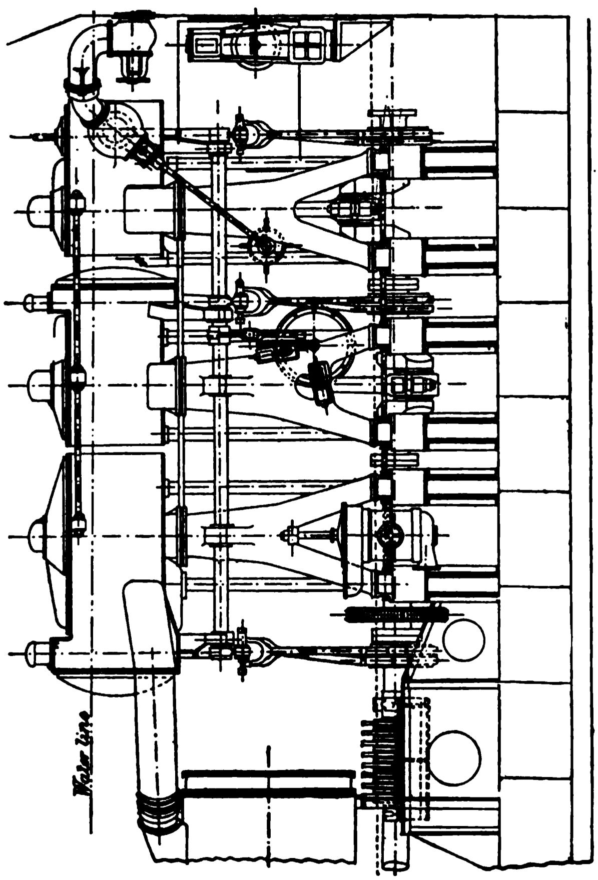


FIG. 11.

The marine steam turbine.—An important development of marine engineering has taken place within the last few years by the practical success of the marine steam turbine in sea-going vessels. This propelling engine, the introduction and success of which are due to the genius of the Hon. C. A. Parsons, is of the rotary variety and has no reciprocating parts. It has been fitted in torpedo-boat destroyers, cruisers, battle-ships, cross-channel steamers, and Atlantic and other mercantile marine steam-ships, and is being applied to many other ships of all classes. It has proved to be very economical at high proportions of full power, to require less attention and repair to maintain running efficiently, and to be free from engine vibration. Its use in the Navy has enabled the amount of boiler power to be reduced considerably, and all new British warships are being fitted with it. It has still greater possibilities as regards the future, and will be found described in detail later.

Forced draught.—The conditions of service of ships in the Royal Navy render it necessary to provide for the development of high power and speed on special occasions, such as the events of action, chasing, &c., although the greater portion of the work of the ship has to be performed at comparatively low powers. It is therefore desirable in warships to provide special means of forcing the boilers when the full speed is required. Formerly a steam jet in the chimney was used for this purpose, but this wastes much fresh water. In 1882 the system of forming the stokeholds into closed compartments, and keeping them under air-pressure by means of blowing fans was adopted and continues to the present day, with results that are satisfactory, provided only a moderate pressure of air be used, and by this means the steam generating powers of the boilers have been largely increased. Details of the fittings required for this purpose are given in Chapter V. The introduction of *forced draught* has enabled the weight of machinery to be considerably reduced, and the average weight of machinery for the latest modern warships, excluding torpedo craft, is from $1\frac{1}{2}$ to $1\frac{3}{4}$ cwt. per I.H.P. developed with moderate forced draught.

Improvements in economy.—In consequence of the improvements effected, the consumption of coal with the most recent engines is less than one-third that required for the engines generally used before 1860, and the effect on warships of this great reduction of coal expenditure has been twofold :—

a. The increased distance ships are able to steam without exhausting their coal supply has rendered seagoing mastless armour-clad ships possible.

b. The reduced quantity of coal necessary to be carried for the same radius of action has enabled space and weight which would formerly have been required for coal to be devoted to other objects, in order to increase their offensive or defensive powers.

Corresponding benefits have also been derived by the mercantile marine, and high speeds are now attained on ordinary passages which would have been impossible without this progress in economy.

Comparison of early with modern engines.—From this brief sketch a general idea may be formed of the progress that has been made in marine engineering. The machinery of the 'Salamander,' built in 1832, weighed 275 tons, developed 400 I.H.P., and consumed 7 to 8 lbs.

of coal per horse-power. In modern warships, machinery of the same weight would, under moderate forced draught, be capable of developing satisfactorily at least 3,000 I.H.P., with about one-fourth the consumption of coal per horse-power, and the space occupied would be considerably less. Another important feature is the great increase in the total power now available for the propulsion of vessels at high speeds. For example, in H.M.S. 'Terrible,' which in 1845 represented the best type of steam warship of the day, the I.H.P. was less than 2,000, and her speed about 10 knots, while in the present H.M.S. 'Terrible,' a first-class cruiser, the horse-power is 25,600, and the speed 22·8 knots. In later cruisers the power is still greater, 48,000 horse-power having been developed, and over 26 knots attained in the British cruisers of the 'Invincible' class, while the two new Cunard Company's large Atlantic steamships 'Lusitania' and 'Mauretania' develop over 60,000 horse-power, and have attained speeds over the Atlantic passage of over 26 knots.

Future progress in steam engines.—Probably the near future will see a further improvement in economy, due to the successful use of superheated steam, especially in conjunction with the turbine engine, the more extended use of which may confidently be anticipated.

Considerable progress is still possible as regards the boiler, in the reduction of the waste of heat which now takes place, due either to incomplete combustion, or the inability of the heat-absorbing surfaces, as now arranged, to prevent a serious loss of heat in the escaping gases.

Internal combustion engines.—The possibilities of the internal combustion engine, in conjunction with a 'producer' for the requisite gas generated from coal, are also being recognised and experimented with, while great strides have recently been made in the design of liquid fuel engines, using oil sprayed direct into the working cylinders. Several examples of these are now in successful operation, while others of large size are under construction. A considerable gain in economy is thus possible, so that as practical difficulties are overcome, the steam engine for propulsion of ships will have a powerful rival.

CHAPTER II.

WORK AND EFFICIENCY.

Force, work, and energy.—Force is that which acts in producing or resisting motion in a body, and may be represented by a pressure or a pull. The British unit of force is the weight of one pound avoirdupois, and forces are therefore expressed generally as being equal to so many pounds weight.

A force is said to perform work when by its action resistance is overcome and motion produced. This union of force and motion is essential to the conception of *work*. However great the pressure applied, unless the body acted on be moved, no work is done. *Energy* is the term used to signify the capacity of a body for doing work. For example, if a force acts through a certain distance it is said to *exert energy*, while the resistance overcome through a certain distance by means of this exertion is the *work done*.

Measurement of work and energy.—The amount of work done is measured by multiplying the magnitude of the resistance—or, in other words, the force opposing the motion—by the distance through which the resistance is overcome, estimated in the direction of the resistance. Energy is measured in a similar manner.

The British unit of work is the foot-pound, which is a very convenient term, implying the combination of force and motion, which is the essential condition for the performance of work. One foot-pound represents the amount of work done in raising a weight of one pound through a distance of one foot, or more generally the exertion of a pressure of one pound through the distance of one foot. If 20 pounds be raised 50 ft., the amount of work performed is represented by $20 \times 50 = 1{,}000$ foot-pounds.

Sometimes for convenience other units of work are used, but they are all formed on the same basis and expressed in a similar manner. For example, the work performed in raising one ton one inch is sometimes called one inch-ton, and it is equal to 2,240 inch-pounds or $\frac{2240}{12}$ foot-pounds. The work of lifting one ton one foot is one foot-ton, and so on. It will be seen that the different terms used are self-explanatory and are convertible one to another. The foot-pound is, however, the general unit, the others only being employed for convenience in special cases.

Power, horse-power.—In the conception of work and energy no question of time enters. When, however, we consider also the time taken to perform so much work, we are considering *power*. Just as the term *work* necessarily involves distance, so does the term power

involve time as well as distance. *Power* may be defined as the *rate at which work is done.* The natural unit of power would be the power of doing work at the rate of one foot-pound per minute, but it is too small to be convenient in engineering. The unit of power adopted is the power of doing work at the rate of 33,000 foot-pounds per minute. This unit of power is termed a *horse-power.*

Efficiency.—In every machine there are always certain causes acting that produce waste of work, so that the whole work done by the machine is not usefully employed, some of it being exerted in overcoming the friction of the mechanism, and some wasted in various other ways. The fraction representing the ratio that the useful work done bears to the total energy exerted on the machine is called the *Efficiency of the machine*; or

$$\text{Efficiency} = \frac{\text{Useful work done}}{\text{Total energy expended}}.$$

Efficiency of the propelling machinery.—In the case of the propelling machinery of a vessel, the useful work done is measured by its effect in causing the ship to move through the water at a certain speed. The source from which this work is derived is the heat energy contained in the coal burnt in the boiler furnaces, and there are four principal stages in the conversion of this heat energy into useful work, in each of which a certain amount of energy is wasted, involving a loss of efficiency. Any device or arrangement which reduces the amount of energy wasted in any one of these four stages will improve the total efficiency by increasing the useful work done for a given expenditure of fuel; thus it is a matter of some consequence to carefully analyse the losses which occur at each stage, in order that the important sources of loss may be correctly located and, if possible, reduced.

With this object in view we will now consider, separately, each of the stages in the conversion of the heat energy of the coal into useful work, estimate the efficiency of conversion at each stage, and enumerate the principal causes of loss of energy.

The first stage in the conversion occurs in the boilers, where the heat evolved by the combustion of the coal generates steam. The second stage takes place in the engine cylinders, where the heat and pressure energy contained in the steam is converted into mechanical work on the pistons of a reciprocating engine or the revolving blades of a steam turbine. In the following paragraphs we shall confine ourselves to the reciprocating engine. The third stage is that in which, by means of the mechanism employed, the reciprocating motion of the pistons is transformed into rotary motion of the propeller, and the energy communicated by the steam to the pistons in the engine cylinders is transmitted to the propeller. The fourth stage is that in which, by means of the action of the propeller, the energy transmitted to it is converted into useful work in propelling the vessel.

Efficiency of the boiler.—In the first stage, only a portion of the heat contained in the coal supplied to the boiler furnaces is transmitted to the water contained in the boilers. In consequence of the heat so transmitted, the feed-water is raised to a temperature corresponding to the steam pressure in the boilers, and is converted into steam at that temperature. Hence, each pound of steam leaving the boilers carries away with it a certain amount of heat in excess of that which it con-

tained when it entered the boiler as feed-water ; and the ratio which the excess heat carried away bears to the heat of combustion of the coal used in supplying this heat, is defined as the *efficiency of the boiler*, and may be expressed either as a fraction or as a percentage. Thus, in saying that a boiler has an efficiency of 70 per cent., it is meant that the difference between the number of thermal units contained in one pound of steam on leaving the boiler and the number of thermal units contained by one pound of feed-water on entering the boiler is 70 per cent. of the number of thermal units contained in the coal burnt in generating this one pound of steam.

Tables are given in Chapter IV. of actual efficiencies of boilers of different types working under different circumstances. It varies considerably as will be seen, but an ordinary value is 70 per cent., sometimes as in torpedo boats it is much less than this or about 60 per cent., while in special cases in the mercantile marine it rises to over 80 per cent.

Efficiency of the steam.—Secondly, the steam generated in the boilers passes along the main steam pipes and enters the main engine cylinders, where some of the heat and pressure energy which it possesses is transformed into work on the pistons. Owing principally to the narrow limits of temperature between which the engine is worked, and to the large amount of heat wastefully rejected to the circulating water in the condensers, only a small proportion of the energy contained in the steam appears as indicated work on the pistons. This is by far the largest loss in any of the four stages in the conversion of heat into useful work. The ratio of the work done on the pistons in a given time (as measured by indicator diagrams) to the energy contained in the steam passing to the engines during this time is called the *efficiency of the steam*, and may be expressed either as a fraction or a percentage.

The efficiency of the steam varies very considerably, depending upon the type of engine, the working steam pressure, ratio of expansion, etc. Ordinary types of modern naval engines (using steam of 200 to 250 pounds per square inch pressure, and with a vacuum of from 25 to 28½ inches in the condensers) use from 14 to 18 pounds of steam per I.H.P. per hour, which corresponds to steam efficiencies ranging from 16½ to 12½ per cent. Detailed information respecting this is given in Chapter XI.

Efficiency of the mechanism.—Thirdly, of the mechanical work done by the steam on the reciprocating pistons, only a certain proportion is transmitted to the propeller, the remainder being wasted in friction at the engine bearings.

The ratio which the work available at the propeller bears to the work done on the pistons is called the *efficiency of the mechanism*, or *mechanical efficiency*. It may be taken generally that the efficiency of the mechanism for naval marine engines, with bearings in good adjustment and shafting in line, ranges from 85 to 90 per cent. Detailed information respecting this is given in Chapter XXIII.

Efficiency of the propeller.—Fourthly, only a portion of the energy transmitted to the propeller is usefully employed in propelling the ship, the remainder is dissipated in the sternward energy of the water acted on by the propeller, in blade friction, in wave-making and eddy-

making, in a manner which is very complex and imperfectly understood. The *efficiency of the propeller* is the ratio of the work usefully employed in propelling the ship to the energy available at the propeller. An average value of the propeller efficiency for modern war vessels with modern types of propeller is from 65 to 70 per cent., the latter figure often being reached. See Chapter XXV.

Resultant efficiency of the propelling machinery.—The resultant efficiency of the propelling machinery is made up of the four efficiencies previously mentioned, and is given by the product of the four factors representing respectively the efficiencies of the boilers, the steam, the mechanism, and the propeller. Any improvement in the efficiency of the propelling machinery, and consequently in the economy of its performance, is therefore due to an increase in one or more of these elements, and later we shall deal with these several points, and in each case describe the efforts that have been made to increase the efficiency which in all types of marine steam engine will be seen to be very low.

Example from naval reciprocating machinery.—As an example of what may be regarded as the highest efficiency for naval propelling machinery, cases will be selected from trial results, where each of the four component efficiencies were respectively highest.

These are as follows :—

—	At Full Power	At ¾ Power	At ½ Power
Boiler efficiency . . .	77·8%	81·0%	80·0%
Steam efficiency . . .	15 1%	16·4%	15·9%
Assume mechanical efficiency	90 per cent. in all cases.		
Assume propeller efficiency .	70 per cent. in all cases.		

Hence, the resultant efficiencies of the propelling machinery are, respectively :—

$$\textit{At Full Power} = \frac{77\cdot8}{100} \times \frac{15\cdot1}{100} \times \frac{90}{100} \times \frac{70}{100} = 7\cdot4 \text{ per cent.}$$

$$\textit{At Three-quarter Power} = \frac{81}{100} \times \frac{16\cdot4}{100} \times \frac{90}{100} \times \frac{70}{100} = 8\cdot4 \text{ per cent.}$$

$$\textit{At Half Power} = \frac{80}{100} \times \frac{15\cdot9}{100} \times \frac{90}{100} \times \frac{70}{100} = 8\cdot0 \text{ per cent.}$$

Consequently we see that for naval machinery of the most modern type, the efficiency of the propelling machinery rarely exceeds 7½ to 8½ per cent.

Example from mercantile machinery.—As an instructive comparison let us take the special case of a vessel in the mercantile marine, where elaborate fittings were provided with a view to reducing coal consumption. In this case the coal consumption per I.H.P. per hour was found to vary from 1·07 lbs. on trial to an average of 1·27 lbs. on actual service. The calorific value of the coal was not determined, neither were the mechanical or propeller efficiencies, but assuming that the respective figures were : 12,000 British thermal units per lb. for

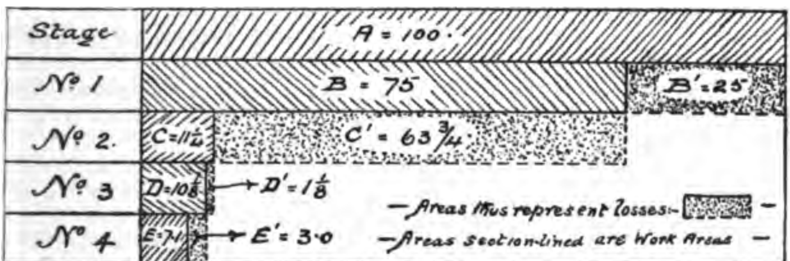
calorific value of the fuel, 90 per cent. for the mechanical efficiency, and 70 per cent. for the propeller efficiency, we see that on trial the resultant efficiency was about:

$$\frac{33000 \times 60}{1{\cdot}07 \times 12000 \times 778} \times \frac{90 \times 70}{100 \times 100} = 12{\cdot}5 \text{ per cent.},$$

which on service was reduced to about 9·85 per cent.

Graphical representation.—Figure 12A represents graphically by the shaded areas the respective losses etc. in the four stages of the conversion of heat into useful work, for a case where:

Boiler efficiency	is taken	= 75	per cent.
Steam efficiency	,,	= 15	,,
Mechanical efficiency	,,	= 90	,,
Propeller efficiency	,,	= 70	,,



Area A (= 100) = Work equivalent to heat value of 1 lb. of fuel.
,, B (= 75) = Work equivalent to heat added to feed water in boiler per pound of fuel burnt.
,, C (= 11¼) = Work done on pistons by steam generated by 1 lb. of fuel.
,, D (= 10⅛) = Work available at propeller per lb. of fuel burnt.
,, E (= 7·1) = Work expended in propelling ship per lb. of fuel burnt.
,, B' (= 25) = Work equivalent to boiler heat losses per lb. of fuel burnt.
,, C' (= 63¾) = ,, ,, engine ,, ,, ,,
,, D' (= 1⅛) = Work lost in bearing friction per lb. of fuel burnt.
,, E' (= 3·0) = Work wasted by the propeller ,, ,, ,,

FIG. 12A.—Graphic Representation of Efficiencies.

CHAPTER III.

HEAT AND ITS EFFECT ON WATER.

In order to comprehend the principles on which the construction and performance of the steam-engine depend, and the object of the various improvements that have from time to time been introduced, it is necessary that the true nature and properties of heat should be known. We will, therefore, as concisely as possible, state the principal points relative to this subject, in order that the succeeding chapters may be clearly understood.

Temperature.—The temperature of a body may be defined as the extent to which it may be capable of communicating sensible heat, or heat that may be felt, to other adjacent bodies.

When two bodies of different temperatures are placed in contact with each other, it is a well-known fact that the hotter body becomes cooler and the colder body hotter, till at length the two bodies become of the same temperature, after which no change in the temperature takes place. This is caused by the passage of heat from the hotter to the colder body, and shows clearly that heat is something that can be transferred from one body to another, so as to diminish the amount of heat in the former body and increase it in the latter.

Effect of and nature of heat.—When heat is added to or abstracted from a body, one of the two following effects is produced: either the temperature of the body is altered, or its state is changed. For example, if heat be added to water under the atmospheric pressure, the temperature is increased until it reaches 212° Fahr. After this the addition of heat does not further increase the temperature, but causes the water to evaporate and become steam—that is, it changes the condition from that of a liquid to that of a gas. Again, if heat be abstracted from water, the temperature is reduced till it reaches 32° Fahr., after which the diminution of heat does not further decrease the temperature, but changes the condition of the water from the liquid to the solid state, forming ice. The quantities of heat passing from one body to another can thus be estimated by the effects produced, so that it is clear that heat is something that can be both transferred and measured.

The true nature of heat has been determined by experiments on friction. It is a matter of common observation that the work expended in friction is apparently lost—that is, it appears no longer in the form of mechanical work; but at every place where friction occurs, heat is developed, and the greater the friction the greater is the amount of heat produced. Experiments have shown that the amount of heat generated by friction is exactly equivalent to the

amount of work lost, and we therefore infer that heat is of the same nature as mechanical work—that is, it is one of the forms of energy.

British thermal unit.—The unit by which heat is measured is called a thermal unit, and in British measurements represents the quantity of heat necessary to raise one pound of water at its maximum density, which corresponds to a temperature of about 39° Fahr., through one degree Fahr.

Joule's equivalent.—The honour of determining the relation between heat and mechanical work belongs to Mr. Joule, who proved, by an elaborate series of careful experiments on the friction of oil, water, mercury, and other substances, that one thermal unit is equal to a definite number of foot-pounds of mechanical work. This number, originally fixed by Joule at 772, is now known by later investigation to be 778—that is, the quantity of heat necessary to raise the temperature of one pound of water at its maximum density, one degree Fahr., can be made to perform work equal to the raising of 778 lbs. one foot high. In honour of the discoverer of the law, this important constant, 778, expressing the relation between heat and mechanical work, is called Joule's equivalent, and is frequently denoted by the letter J.

The convertibility of heat and work, in a definite ratio, is expressed in the following statement, generally known as the mechanical theory of heat, viz.: Heat and mechanical energy are mutually convertible, and heat requires for its production, and produces by its disappearance, mechanical energy in the proportion of 778 foot-pounds for each unit of heat. This statement forms also the *first law of the science of thermo-dynamics.*

Communication of heat.—Heat may be communicated from one body to another in three different ways, viz., by radiation, conduction, and convection.

Radiation.—Radiant heat is given off from hot bodies in straight lines, and the rays of heat are subject to the same laws as the rays of light. As far as the generation of steam is concerned the useful radiation is confined to the furnace, the crowns and sides of which, intercepting the rays of heat from the burning fuel, become themselves heated, and the heat passes through them to the water in the boiler. A considerable amount of heat is given off by radiation from burning coal, and it is very important, therefore, to intercept this, and to insure that as far as possible the whole of the heat diffused in this way should be transmitted, either directly or indirectly, to the water in the boiler, and not wasted on the external air or other bodies.

Radiation is an important item to be considered with reference to the economical employment of steam, for it always causes a certain loss of heat, and unless proper precautions are taken this loss may become very considerable.

The surfaces of the boilers, steam-pipes, cylinders, &c., when the engines are at work, are very much hotter than the surrounding bodies, and consequently, in order that loss of heat by radiation may be avoided as far as possible, all those surfaces should be clothed with some non-conducting material. Hair-felt has been largely employed for this purpose, and this is usually kept in its place by an outer

covering of canvas, wood, or sheet-iron. Preparations of cork and other non-conducting materials have also been used. These substances, however, when applied to very hot surfaces, are in danger of being burnt away, and various incombustible non-conductors, such as asbestos, silicate cotton, fossil meal, &c., are now used. The efficient clothing of the hot surfaces is of great importance, and if it be neglected the economical working of the machinery may be seriously impaired.

Conduction.—The second way in which heat may be transferred from one body to another is by conduction. There are two kinds of conduction, called respectively internal and external conduction. The conduction that takes place between the contiguous particles of one continuous body is called internal conduction. The term external conduction is used when heat passes through the points of contact of two distinct bodies.

In boiler plates and flues the resistance offered to heat entering and leaving the surfaces of the plates is in general so much greater than the resistance offered to its passage through the body of the plate, that the nature and thickness of the plate have little effect on the rate of conduction through it, so that the rate depends on the difference of temperature of the fluids on the two sides of the plate.

The following approximate rule was given by Rankine for the rate of conduction through boiler plates and flues, but its accuracy is somewhat doubtful:—

$$q = \frac{(T_1 - T_2)^2}{a},$$

where q = rate of conduction through the plate, in thermal units per square foot of surface per hour;

T_1 and T_2 = the temperatures on the opposite sides of the plate;

and a = a constant, which is in ordinary cases between 160 and 200.

Convection.—The third method of transfer of heat is by convection. This is the way in which gases and liquids are heated. Conduction, in the true sense of the word, is very slow in liquids, and almost, if not wholly, inappreciable in gases. When heat is applied to the bottom of a vessel containing a fluid, the particles in contact with the bottom are first heated, and become less dense and therefore rise through the superincumbent mass of fluid, allowing cooler particles to take their place, which become themselves heated, and rise and circulate through the mass in a similar manner.

It is essential that circulation and mixture of all the particles of a fluid should take place to cause the temperature to be uniform throughout the mass. In order that heat may be efficiently transmitted through boiler plates and flues, each of the fluids in contact with them—viz. the water on the one side, and the heated gases on the other—should have free circulation, so that the particles in contact with the plates should not be considerably different in temperature from those at some distance from the plates. Boilers are sometimes fitted with circulating plates to set up currents in the water, and with bafflers and bridges in the flues to break up the currents of hot gas and form eddies, in order to promote circulation and mixture in the respective fluids.

It can easily be proved that for the most efficient transfer of heat from a fluid on one side of a plate, to another fluid on the other side, the least difference of temperature between the two sides of the plate should differ as little as possible from the greatest difference of temperature, so that the hottest parts of the two fluids should be adjacent to each other, and therefore the coolest parts will also be adjacent. Consequently the surface of a condenser will be most efficient when the cold condensing or circulating water enters the condenser at the end where the condensed steam leaves it, so that the entering steam gives up its heat to the heated circulating water leaving the condenser.

Similarly in a boiler the best arrangement from this point of view would be that in which the general motion of the entering feed water is in the opposite direction to the motion of the hot gases.

Application of heat to water.—We will now consider the effects produced by the application of heat to water. At first the temperature of the water is raised. The particles of water in contact with the heating surface, which in a marine boiler consists of the plates of furnace, combustion chamber, and tubes, become heated, and rise and circulate through the mass of water, their places being taken by cooler particles, till at length the whole of the water is raised to the boiling point by the convection of heat.

Sensible heat.—The heat added to the water up to the temperature at which boiling occurs is generally called sensible heat, its effect being simply to change the temperature, and not the state, of the water, and its amount may be calculated by means of the thermometer.

Latent heat.—After this, in order to convert the boiling water into steam, a large quantity of heat has to be expended which does not produce any increase in the temperature. This heat is known as *latent heat.*

During the period when heat was considered as a kind of substance called *caloric*, it was supposed that the quantity of heat required for evaporation became hidden or latent in some way during the change from the liquid to the gaseous state, and that it again became sensible or tangible on the reverse process being performed. We now, however, know that heat is not a substance, but simply one of several forms of mechanical energy, and the development of the science of thermodynamics has shown that this amount of heat, instead of being lost or hidden, is simply expended, principally in performing the work of overcoming the molecular cohesion of the particles of water which resists the change of state, and also in overcoming the resistance of external bodies to the change of volume which ensues. Work is therefore done in changing the water from a liquid into a gas, and this is stored up as mechanical energy, which can be yielded back again either as work, or heat, when the gas returns to the original state of water.

The term *latent heat* has been retained for the sake of convenience, but it must be understood as an expression that means simply *the quantity of heat that must be added to or subtracted from a body in a given state, to change it into another state without altering its temperature.*

Boiling point.—The boiling point, or the temperature of ebullition of any liquid, may be defined as that stage in the addition of heat to

the liquid at which the pressure on it is just overcome by the pressure of vapour due to the temperature.

The temperature of the boiling point depends on the *pressure* under which the liquid is evaporated. The greater the pressure the higher is the temperature at which the liquid boils.

Pressure in lbs. per square inch by gauge	Pressure in lbs. per square inch absolute	Temperature of boiling point Fahrenheit	Latent heat in B.T.Us.	Sensible heat from 32° F., B.T.Us.	Total heat from 32° F., B.T.Us.	Volume of 1 lb. of steam in cubic feet	Weight of 1 cubic foot of steam in lbs.	Increase of pressure in lbs. per sq. in. for difference of 1° F.	Increase of temperature in degrees Fahr. for increase of 1 lb. pressure
1	2	3	4	5	6	7	8	9	10
−4·7	10·	193·24	979·0	161·91	1140·91	37·84	·0264	·250	3·991
0	14·7	212·00	965·7	180·90	1146·60	26·36	·0380	·398	2·510
15	29·7	249·65	938·9	219·15	1158·05	13·62	·07343	·542	1·846
20	34·7	258·88	932·3	228·55	1160·86	11·78	·08492	·644	1·554
25	39·7	266·65	926·8	236·47	1163·30	10·371	·09641	·693	1·440
30	44·7	273·87	921·7	243·86	1165·50	9·280	·10777	·758	1·320
35	49·7	280·47	917·0	250·60	1167·50	8·396	·11908	·824	1·210
40	54·7	286·54	912·6	256·76	1169·36	7·677	·13028	·888	1·128
45	59·7	292·18	908·5	262·56	1171·07	7·080	·14143	·948	1·056
50	64·7	297·46	904·6	268·07	1172·66	6·557	·15258	1·029	·972
55	69·7	302·32	901·2	273·00	1174·20	6·108	·16354	1·046	·956
60	74·7	307·10	897·7	277·89	1175·60	5·729	·17447	1·131	·888
65	79·7	311·54	894·5	282·44	1176·95	5·393	·18539	1·183	·844
70	84·7	315·76	891·5	286·78	1178·24	5·096	·19632	1·242	·804
75	89·7	319·78	888·6	290·90	1179·45	4·825	·20724	1·295	·772
80	94·7	323·64	885·8	294·89	1180·62	4·587	·21801	1·351	·740
85	99·7	327·34	883·1	298·64	1181·74	4·371	·22876	1·408	·710
90	104·7	330·89	880·5	302·37	1182·84	4·171	·23951	1·457	·686
95	109·7	334·32	878·0	305·90	1183·89	3·996	·25026	1·520	·658
100	114·7	337·61	875·6	309·28	1184·87	3·836	·26091	1·597	·626
110	124·7	343·87	871·1	315·73	1186·78	3·543	·28221	1·698	·589
120	134·7	349·76	866·8	321·86	1188·64	3·296	·30351	1·795	·557
130	144·7	355·33	862·7	327·63	1190·34	3·077	·32470	1·905	·525
140	154·7	360·58	858·9	333·08	1191·94	2·893	·34577	1·996	·501
150	164·7	365·59	855·2	338·28	1193·47	2·726	·36684	2·096	·477
160	174·7	370·36	851·7	343·22	1194·92	2·577	·38791	2·188	·458
170	184·7	374·94	848·3	347·98	1196·29	2·448	·40871	2·288	·437
180	194·7	379·31	845·1	352·54	1197·61	2·330	·42951	2·381	·420
190	204·7	383·51	842·0	356·86	1198·90	2·222	·45006	2·463	·406
200	214·7	387·57	839·0	361·14	1200·14	2·126	·47062	2·558	·391
210	224·7	391·48	836·1	365·23	1201·33	2·036	·49118	2·653	·377
220	234·7	395·25	833·3	369·16	1202·48	1·954	·51174	2·732	·366
230	244·7	398·91	830·6	372·96	1203·59	1·879	·53230	2·825	·354
240	254·7	402·45	827·9	376·65	1204·58	1·809	·55279	2·907	·344
250	264·7	405·89	825·4	380·21	1205·64	1·744	·57319	2·994	·334
260	274·7	409·23	823·0	383·68	1206·68	1·685	·59358	3·077	·325
270	284·7	412·48	820·6	387·11	1207·71	1·629	·61399	3·165	·316
280	294·7	415·64	818·2	390·46	1208·67	1·576	·63440	3·247	·308
290	304·7	418·72	815·9	393·75	1209·61	1·528	·65485	3·322	·301
300	314·7	421·73	813·6	396·88	1210·54	1·481	·67516	3·401	·294
310	324·7	424·67	811·4	399·99	1211·45	1·438	·69545	3·484	·287
320	334·7	427·54	809·2	403·10	1212·34	1·398	·71574	3·571	·280
330	344·7	430·34	807·1	406·10	1213·18	1·359	·73605	3·650	·274
340	354·7	433·08	805·0	409·01	1213·98	1·322	·75636	3·731	·268
350	364·7	435·76	802·9	411·84	1214·77	1·288	·77662		

The foregoing table gives the boiling temperatures of fresh water for various pressures, ascertained from the experiments of M. Regnault.

The temperature depends on the *total or absolute pressure*, i.e. the pressure including that of the atmosphere, so that in ascertaining the temperature corresponding to any given pressure, as shown on an ordinary pressure gauge, the atmospheric pressure must be added.

To obtain the latent heat corresponding to any pressure intermediate to those given above, it will be sufficient for practical purposes to use the method of interpolation, thus: to ascertain the latent heat corresponding to 106 lbs. by gauge we have:—Latent heat for 100 lbs. = 875·6 B.T.U. Difference for 10 lbs. = 4·5 $\therefore$ Difference for 6 lbs. = $\frac{6}{10} \times 4{\cdot}5 = 2{\cdot}7$ $\therefore$ Latent heat at 106 lbs. = 872·9 B.T.U. Similar methods may be used for intermediate values in columns 5 to 8. Intermediate values for corresponding pressures and temperatures may be obtained by the use of columns 9 and 10.

To meet the cases of pressures below atmospheric pressure the following table has been prepared showing the temperature corresponding to various amounts of vacuum. It will be found useful in dealing with questions concerning condensers:—

Vacuum measured in inches of Mercury	Absolute pressure in inches of Mercury	Absolute pressure in lbs. per square inch	Temperature of boiling point in degrees Fahrenheit	Latent heat of evaporation in B.T.Us.	Sensible heat of evaporation from 32° F. in B.T.Us.	Total heat of evaporation from 32° F. in B.T.Us.
Gauge						
29½	½	·245	59·1	1072·8	27·1	1100·0
29	1	·490	79·3	1058·8	47·3	1106·1
28½	1½	·735	92·0	1049·9	60·1	1110·0
28	2	·980	101·4	1044·4	69·5	1112·8
27	3	1·470	115·3	1033·7	83·4	1117·1
26	4	1·96	125·6	1026·5	93·8	1120·3
25	5	2·45	134·0	1020·6	102·2	1122·8
24	6	2·94	141·0	1015·7	109·3	1125·0
23	7	3·43	147·0	1011·5	115·3	1126·8
22	8	3·92	152·3	1007·8	120·5	1128·4
21	9	4·41	157·0	1004·5	125·4	1129·8
20	10	4·90	161·5	1001·3	129·9	1131·2
19	11	5·39	165·6	998·4	134·1	1132·4
18	12	5·88	169·2	995·9	137·7	1133·5
17	13	6·37	172·8	993·4	140·3	1134·6
16	14	6·86	176·0	991·1	144·5	1135·6
15	15	7·35	179·1	988·8	147·7	1136·5
14	16	7·84	182·0	986·9	150·6	1137·4
12	18	8·82	187·4	983·1	156·0	1139·1
10	20	9·80	192·3	979·6	161·0	1140·6
5	25	12·25	203·0	972·1	171·8	1143·9
0	30	14·70	212·0	965·7	180·9	1146·6

14·7 lbs. = atmospheric pressure = 30 inches of mercury.

The boiling point of a liquid is also affected by its *density*. Solid matter dissolved in the liquid, as, for example, salt in water, resists ebullition and increases the temperature at which the liquid boils.

Ordinary sea-water contains about $\frac{1}{32}$ or $\frac{1}{33}$ part of solid matter, and this raises the temperature of the boiling point by 1·2° Fahr., so that the boiling point of sea-water under the atmospheric pressure, instead of being 212° Fahr., is 213·2° Fahr. The density of the water in marine boilers in ordinary work is generally not allowed to exceed three to four times that of sea-water, and at this density the boiling point at atmospheric pressure would be about 216° Fahr.

Total heat and latent heat of evaporation.—The *total heat of evaporation* is the sum of the sensible heat and latent heat of evaporation, and is defined as the quantity of heat necessary to raise one pound of water from the freezing point, 32° Fahr., to a particular temperature, and to evaporate it at that temperature.

If H represent the total heat of evaporation, L the latent heat of evaporation, and S the sensible heat, we have

$$H = L + S.$$

The *latent heat of evaporation* of a pound of steam in thermal units at any given temperature of evaporation, t, is given by the following approximate formula :—

$$L = 966 - 0{\cdot}7\ (t - 212)$$

Though the latent heat diminishes considerably as the temperature of evaporation is increased, the increase of temperature, or, in other words, the increase of sensible heat, is greater than the decrease of the latent heat, so that the *total heat of evaporation* is gradually increased. This is shown in column 6 of table.

The table also gives approximately, in thermal units, the latent, sensible, and total heat of evaporation of one pound of steam up to a pressure of 350 lbs. per square inch above the atmosphere, which is not exceeded by the highest pressure used at present in marine boilers, whether cylindrical or water-tube. The volume of one pound of steam and the weight of one cubic foot have also been added for each pressure.

The *total heat of evaporation*, at any temperature T, may be calculated from the following formula [1] :—

$$\begin{aligned} H &= L + S \\ &= 966 - 0{\cdot}7\ (t - 212) + (t - 32) \\ &= 1082 + 0{\cdot}3t \end{aligned}$$

This formula shows very clearly that the rate of increase in the total heat of evaporation, as the temperature of evaporation is raised, is about $\frac{3}{10}$ of a thermal unit for each degree of rise of temperature.

Evaporation of water under constant pressure.—The different stages in the evaporation of water just discussed may be summarised in the following manner :—

Suppose one pound of water, at a temperature of 32° Fahr., to be

[1] These formulæ assume the specific heat of water to be constant, which assumption is sufficiently accurate for the purpose of this book. It is found, however, that the specific heat gradually increases, so that at a temperature of 400° F., it is about 4 per cent. greater than it is at 39° F.; allowing for this, more accurate formulæ are

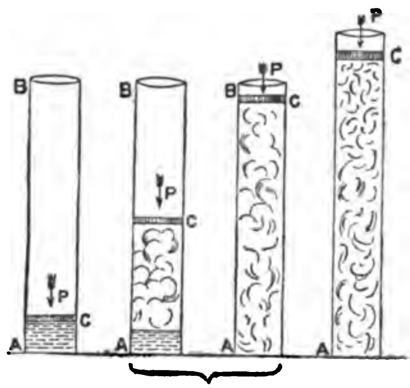
$$L = 966 - {\cdot}71\ (t - 212) \text{ and } H = 1082 + {\cdot}305\ t$$

contained in a cylinder A B, open at the top, in which a piston C works steam-tight, loaded with a certain weight. The pressure on the water, indicated by arrows, consists of the weight of the piston, the weights added, and the atmospheric pressure. This pressure is represented by P pounds per square inch on the piston.

In Fig. 13 three stages in the process of applying heat to the bottom of the cylinder are shown.

Practically no movement of the piston occurs until the temperature of evaporation is attained, which temperature will depend on the amount of the pressure P produced by the piston on the water. This part of the process, viz. the raising of the temperature from the freezing to the boiling point, is shown in Stage 1.

From this point the *temperature* remains constant, and steam is given off at the constant pressure P pounds per square inch, which causes the piston to gradually rise, as shown in the two figures of Stage 2, the piston continuing to rise during the addition of heat, as more and more of the water is evaporated, until the whole of the water is turned into steam. The first sketch for Stage 2 represents the condition of affairs when only a portion of the water has been evaporated; the second sketch for this stage represents the final condition, when all the water has just been evaporated. During the whole of this stage the temperature remains constant at that corresponding to the pressure of evaporation P.



Stage 1. Addition of sensible heat. Rising temperature.

Stage 2. Addition of latent heat. Steam saturated. Constant temperature.

Stage 3. Superheating. Rising temperature.

FIG. 13.

The total heat of evaporation H supplies the energy for all these changes, and is expended as follows:—(1) In raising the temperature of the water to boiling point. (2) In converting the water into steam. (3) In doing external work in raising the loaded piston. (4) In raising the centre of gravity of the mass.

It should be carefully kept in mind that of the total heat H supplied, a part, namely, that expended in performing (3) and (4), is transformed into mechanical energy, so that the heat contained in the steam in excess of that originally contained in the water is less than H by the heat equivalent of the work so done. In practice the heat equivalent of (4) is neglected, its amount being so small as to permit this.

If, after all the water is evaporated, more heat be applied to the steam in the cylinder, the pressure P on the piston remaining the same, the volume will continue to increase still further, while the temperature also will again increase, and both volume and temperature would increase without limit if sufficient heat be applied. The steam in this case is said to be superheated, and is indicated in Stage 3.

In Stage 1, therefore, we have increase of temperature without any practical increase of volume; in Stage 2, increase of volume without

increase of temperature; in Stage 3 we have increase of both temperature and volume.

For a complete theory of the steam-engine a thorough knowledge of all these stages would be required, but the properties of steam in the last stage, viz. when heated without being in contact with water, have not yet been fully ascertained, and in investigations on the action of superheated steam it is generally assumed that its properties are sensibly those of a perfect gas. This assumption appears to be correct, provided the steam be considerably superheated, but is not if the superheating is only moderate.

Steam for marine engines which has been superheated by the direct application of heat is not now generally used, but the modern tendency is to re-introduce it, and it has been fitted with success in several cases. The formation of steam which has been superheated to a certain extent indirectly by the action of the mechanism is, however, common, and accurate information as to its behaviour when in this condition seems desirable.

Extensive experiments have however been made, giving almost all necessary information with reference to the other stages, which are those that most concern us in this treatise.

Evaporation of water in a vessel of constant volume.—An important modification in the conditions of steam generation in the above example will now be considered. It has been pointed out that the piston in the cylinder illustrated above, is free to move and allow the steam to gradually expand in volume as the water is evaporated, but suppose that this is not the case, and that the piston is fixed in the cylinder at a certain distance from the water, so that whatever steam is formed from the water, the volume of steam and water remains constant, and is represented by A C in Fig. 14. Suppose further that only the water occupies the space A C, so that the absolute pressure therein is zero. If heat be now applied to the water, instead of the formation of steam taking place only when a certain temperature is reached, as in our first case, steam immediately commences to be formed, and *both the temperature and the pressure* gradually increase as more of the water is evaporated, although the temperature and pressure are still connected by the same law as before.

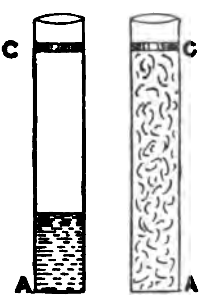


FIG. 14.

The pressure and temperature of the steam when all the water has just been evaporated can be ascertained from our knowledge of the weight of water, and the known volume of the steam, for the weight of steam is the same as that of the water from which it was formed, so that its volume per pound can be calculated, whence from column 7 of table, page 26, its pressure and temperature can be ascertained.

If the vessel be again assumed to have originally contained 1 lb. of water, and A C to be of the same volume as in the final condition of Stage 2 (Fig. 13), then at the moment of its complete evaporation the condition of the steam in the two vessels will be identical, and consequently the amount of heat contained in it will be the same in each case. If W be the heat equivalent of the work done in

raising the piston in Stage 2, H-W will be the heat in the steam in excess of that originally contained in the water.

If V be the volume of the pound of steam in cubic feet and P the pressure on the piston in lbs. per sq. ft., it can be readily shown that W is approximately equal to PV foot pounds, or if p be the pressure in lbs. per sq. in., W is equal to 144 p V foot pounds, or, since $\frac{144}{778} = \cdot 1851$, $W = \cdot 1851\ p\ V$ British thermal units, so that the heat in the steam in excess of that contained in water at 32° F. is

$$\begin{aligned} H-W &= H - \cdot 1851\ p\ V \\ &= t-32+966-\cdot 7(t-212)-\cdot 1851\ p\ V, \end{aligned}$$

or generally if instead of 32°, t_o be the original temperature of the water

$$H-W=t-t_o+966-\cdot 7(t-212)-\cdot 1851\ p\ V$$

and this quantity is usually designated by the letter I, so that we write

$$\begin{aligned} I &= H-W \\ &= t-t_o+966-\cdot 7(t-212)-\cdot 1851\ p\ V. \end{aligned}$$

If only a fraction x of the pound of water in Stage 2 had been evaporated, then the heat of evaporation is designated for the sake of distinction by the letter h, therefore

$$\begin{aligned} h &= S+x\ L \\ &= t-t_o+x\{966-\cdot 7(t-212)\} \end{aligned}$$

and since the work done in this case is $P \times x$ V foot pounds, or $\cdot 1851\ p\ x$ V British thermal units, the heat contained in the steam and water in excess of that contained in the water from which it was formed, would be equal to $h - \cdot 1851\ p\ x\ V$. Calling this quantity i we have

$$i=t-t_o+x\ \{966-(t-212)-\cdot 1851\ p\ V\}$$

The fraction x is called the dryness fraction, and $1-x$ i.e. the quantity of the pound of water unevaporated, is called the wetness fraction. If in the solution of any problem we find $x=1$, we know that the steam is dry and saturated; if less than 1, that it is wet, and if greater than 1, it follows that the volume is greater than that corresponding to the evaporation of all the water, so that the steam is superheated.

Heat required to raise steam in a boiler.—The case described at Fig. 14 occurs in practice when steam is being raised in a boiler, from cold water, with the stop-valves and other outlets shut, except that instead of there not being any pressure on the water prior to the application of heat, there is the pressure of the atmosphere. In this case the temperature of the water will gradually rise, but no steam will be formed, nor will the pressure increase till the temperature reaches that corresponding to the atmospheric pressure—viz. 212° F.—when steam commences to be given off, and both temperature and pressure rise. In our boiler, also, the evaporation does not continue till all the water is evaporated, as the amount of water is so large compared with the volume of the boiler that the desired pressure is attained when only a small portion of the water has been evaporated. To ascertain

exactly the expenditure of heat necessary to accomplish this raising of steam pressure, we note that the whole weight of water has been raised to the final temperature and a small fraction of it evaporated. This fraction will be the x of our last formula, which will then enable us to obtain accurately the amount of heat added.

In an ordinary cylindrical boiler, however, the volume of water is generally from ·65 to ·75 times the whole internal volume of the boiler, so that the amount of heat expended in formation of steam is comparatively very small.

In all such cases of raising steam from cold water, or of raising steam pressure from one point to another, the expenditure of heat in thermal units may, for all practical purposes, be taken as $W(t_1-t_2)$ where

$$W = \text{weight of water in pounds,}$$
$$t_1 = \text{final temperature of water,}$$
$$t_2 = \text{initial temperature of water.}$$

The error due to neglecting the formation of steam will not exceed about 1 per cent., even at high pressures, with the proportions common in the usual marine boilers.

CHAPTER IV.

COMBUSTION AND ECONOMY OF FUEL—BOILER EFFICIENCY.

In this chapter the processes through which fuel passes during combustion will be considered, and the precautions necessary to insure economy in the process.

Owing to the extent of its occurrence in nature and being always easily obtainable, the common fuel for boilers practically consists of coal. Mineral oil is also now used, especially in neighbourhoods near the sources of production, and with much more success than at first. Both these fuels consist essentially of

Carbon,
Hydrogen,
and Oxygen,

and of these, the heating elements are hydrogen and carbon. Sulphur is sometimes present in small quantities, but its heating effect is small.

The presence of oxygen is considered to detract from the value of a fuel, as the oxygen is generally regarded as being already combined with its proportion of hydrogen in the form of water (H_2O). As these gases combine in the proportion of $1:8$ by weight, this virtually reduces the amount of hydrogen available for combustion by one-eighth of the weight of the oxygen present.

Combustion.—Combustion is simply a *rapid chemical union of oxygen with the hydrogen and carbon* in the fuel, heat being evolved during the process. When burning fuel in the furnace of a boiler, the main object to be effected is to obtain this heat and utilise it for the generation of steam. On account of the almost universal adoption of coal as a fuel, its combustion will be considered somewhat at length. In the first place a certain temperature is required for ignition, so that it is necessary to apply heat to start the process, but the subsequent chemical action produces the necessary temperature to continue the combustion.

There are two distinct stages in the combustion of coal, viz. the formation and combustion of the gases, and the combustion of the solid residue. The gaseous constituents of coal are not given off *during* the combustion, as the distillation of the gases is an entirely distinct operation from the combustion of the gases or solid residue.

Consider the combustion of bituminous coal, which is the most difficult to burn economically in furnaces on account of the inefficiency of the ordinary arrangements for consuming the gaseous products.

When coal is thrown on a bright fire, it at first absorbs heat from the fire to liberate the gaseous constituents. These are the same as the coal gas generated in retorts and used for lighting purposes, and

consist principally of light carburetted hydrogen gas (fire-damp or marsh gas), represented by the chemical symbol CH_4 and heavy carburetted hydrogen gas (olefiant gas), denoted by the formula C_2H_4. Until all the gases are separated from the solid part of the fuel no combustion takes place, and the coal remains in an unburnt and comparatively cool state, so that, unless proper precautions are taken to consume the gases, they may be a source of loss instead of gain, because heat is abstracted from the fire to effect their liberation.

On the application of heat the hydrogen and carbon are separated; and by providing an adequate supply of oxygen by means of atmospheric air each enters independently into combination with oxygen, forming steam and carbonic acid gas. From the combining equivalents of hydrogen, carbon, and oxygen, we find that each volume of light carburetted hydrogen gas will require two volumes of oxygen for its combustion, and each volume of the olefiant gas three volumes of oxygen. Thus, between two and three volumes of oxygen will be required for the complete combustion of each volume of the gas formed in the furnace; and as the oxygen in atmospheric air amounts to only one-fifth of its bulk, between ten and fifteen volumes of air will be required for each volume of gas.

Air required for combustion of gases.—One ton of bituminous coal is estimated to produce about 10,000 cubic feet of gas; therefore 100,000 to 150,000 cubic feet of air must actually combine with the gases produced from each ton of coal to effect their complete combustion. To insure thorough mingling of the air with the gas, in order to effect perfect combustion, it is found that from one and a half times to twice the quantity of air theoretically necessary must be admitted; so that a minimum of 150,000 to a maximum of 300,000 cubic feet of air might be required for the combustion of the gases alone produced from one ton of bituminous coal, and it may be taken that not less than 200,000 cubic feet are necessary. All this air should, if the fires are thick, be admitted above the bars, as, if it were allowed to pass through the burning coal on the grate, it would be deprived of a great portion of its oxygen, and its value for burning the gas be depreciated.

Combustion of the solid carbon.—We come now to the coke or carbon that remains on the bars after the gases have been disposed of. The air for its combustion must pass between the bars and through the fuel. At first air is in excess and the union of oxygen with the carbon is complete, and carbonic acid gas (CO_2) is formed; but if the layer of coal is thick relatively to the quantity of air passing through it, much of this gas as it rises through the fire takes up more carbon, forming carbonic oxide gas (CO), and unless arrangements are made to consume this gas by adding additional oxygen, a large quantity of heat will be wasted, the products of combustion passing off as carbonic oxide, by which less than one-third the heat is produced that would be yielded if the combustion were complete and the products passed off in the form of carbonic acid. It is therefore necessary, especially with thick fires, to admit air above the fuel, for the complete combustion of the carbon, in addition to that referred to already as necessary for the combustion of the gases.

Stoking.—The question of firing or stoking will now be considered a little in detail. Coal can easily be thrown through the fire doors, but skill

is necessary to place it on those parts of the grate which require it. This is one of the chief requirements of good stoking, viz. to produce a fire of fairly uniform thickness over its whole area. If any depressions occur, the resistance to the passage of air is reduced so that the air flows through this part in greater volume and the coal is rapidly burned away and the bars may soon become bare.

With thin fires no doubt sufficient air for complete combustion may pass through the fuel, in which case we are independent of the air supply above it. More air also passes through a thin fire and so increases the rate of combustion and power of the boiler, unless carried to extreme limits.

With thick firing, although the doors are opened less frequently, thus causing less labour, large volumes of gases are suddenly produced, for which there is generally not sufficient air for combustion, and an undue lowering of temperature is caused, which may also prevent ignition. This is especially the case in most water-tube boilers, where the time for the gases to mingle with air and burn before reaching the tubes is small. On the whole, the plan giving the best results with usual arrangements is to 'fire lightly, fire quickly, and fire often'; and as this involves the greatest expenditure of manual labour on the part of the stokers, constant watchfulness on the part of persons in charge will be required to see it fully carried out. With large grates, such as most water-tube boilers are provided with, the two halves of the grate should be fired alternately at equal intervals, so that the gases liberated from the coal thrown on one half the grate are kept up in temperature for ignition and combustion, by the glowing carbon of the other half. For similar reasons the furnaces of tank boilers leading into the same combustion chamber should never be fired at the same time, but at intervals of four or five minutes.

Total quantity of air required.—The quantity of air necessary for the combustion of the carbonaceous portions of a ton of coal can be estimated as follows. Every 6 lbs. of carbon requires, in order to form carbonic acid gas (CO_2), 16 lbs. of oxygen. The volume of air necessary to supply this would be about 900 cubic feet. Assuming 80 per cent. of carbon in the coal, about 270,000 cubic feet of air are required *theoretically* for the combustion of the solid residue of each ton of coal after the gases have been distilled.

Increasing this from one and a half times to double the amount, as in the case of the gas, to ensure perfect mingling, we get from 405,000 to 540,000 cubic feet, which, added to the 200,000 cubic feet required for the gas, make a total of 605,000 to 740,000 cubic feet, which enormous volume of air is necessary for the complete combustion of each ton of coal.

The following formula is given by Rankine for the weight of air approximately necessary for the complete combustion of each pound of fuel :—

$$A=12\left\{C+3\left(H-\frac{O}{8}\right)\right\}$$

where A=lbs. of air required per lb. of fuel ; and C, H, and O are the fractions of carbon, hydrogen, and oxygen respectively contained in the fuel.

This formula is obtained from the following considerations. The

average composition of air by weight is practically 77 parts of nitrogen and a few other constituents, and 23 of oxygen; and hence to obtain 1 lb. of oxygen, $\frac{100}{23}$ lbs. of air are required. From their atomic weights, it is seen that for complete combustion 1 lb. of carbon would require $\frac{32}{12}$ lbs. of oxygen, or $\frac{32}{12} \times \frac{100}{23} = 11{\cdot}6$ lb. of air; and 1 lb. of hydrogen would want three times this quantity. Further, the oxygen in a fuel is assumed to be combined with its proportion of hydrogen (see p. 33), and thus the expression

$$A = 11{\cdot}6\left\{C + 3\left(H - \frac{O}{8}\right)\right\}$$

is eventually obtained; and in Rankine's formula quoted, the number 11·6 is replaced by the approximation 12.

By experience in actual cases it is found that where the draught is produced by artificial means, such as a fan, this theoretical quantity must be increased by one-half; and in cases of natural or ordinary chimney draught it must be doubled in order to insure perfect mingling.

In practice it is not necessary to calculate with exactness the quantity of air required, and it is sufficient for all practical purposes to take 12 lbs. of air as the quantity chemically necessary for the combustion of each pound of coal. With natural draught, therefore, 24 lbs. of air must be supplied for each pound of coal to insure perfect admixture and combustion. With artificial draught 18 lbs. of air would probably be sufficient for each pound of coal. This forms, therefore, one advantage of accelerated draught, the temperature of the fire being not reduced so much, so that the radiation of heat from the burning fuel is greater, a higher flame temperature is obtained, and the loss from the heat carried off by the nitrogen in the air and excess oxygen less.

Fire-grate.—The carbon being a solid body requires only a definite space, and since its combustion depends on the amount of air supplied to it, and not on the space it occupies, the area of the grate will depend on the draught employed. In locomotive and other boilers with forced draught the rate of combustion is often as high as from 80 to 90 lbs. of coal per square foot of grate per hour, whilst in those marine boilers with draught due only to the heated gases in the funnel the rate is generally not much more than 20 lbs. per square foot of grate per hour.

It is essential for economy that the length of grate should be kept within such limits that it may be kept well and uniformly covered with coal. With very long grates there is danger that the back parts will not be properly covered, so that volumes of cold air rush in, cooling the gases, tubes, &c., and causing a considerable waste of heat.

Combustion chamber.—In the combustion chamber the gases are combined, and consequently allowance has to be made for their expansion; the space above or beyond the fuel should therefore be made as large as possible. During the last few years the tendency has been to increase the combustion chambers of the tank boilers even at the expense of the loss of a certain amount of tube-heating surface, and this is found generally to increase the economical performance of the boilers. The combustion chambers of large single-ended tank boilers were often not more than 15 to 18 inches deep from the backs

of the chambers to the tube-plates, but in the later examples they are seldom less than from 22 to 30 inches deep, and in many cases even larger. One method of construction that has the practical effect of increasing this size is to lead the furnaces into a common combustion chamber. By firing the furnaces alternately, the gas given off has the whole volume of the common combustion chamber of the furnaces in which to expand, as well as the heat from the adjacent furnaces to prevent the gases from falling below the temperature of ignition.

As regards the influence of combustion chamber space on the efficiency of a boiler, an interesting experiment was carried out on a marine boiler at Devonport by shortening the tubes one foot and adding to the width of combustion chamber by this amount. It was found that the efficiency of the boiler remained exactly as before the change, the greater volume for combustion, and the increased combustion-chamber surface making up for the loss of a much greater area of tube surface. A limit would, however, soon be reached, beyond which this reduction of tube surface could not be made without loss of efficiency.

It should always be borne in mind that no amount of heat can combine the gases unless air be supplied; but, on the other hand, if the gases are not kept up to a certain temperature, called *the temperature of ignition*, the oxygen of the air will not chemically unite with them and cause combustion to take place.

In the water-tube boilers introduced into the Navy, it has not always been found possible to arrange for a corresponding combustion chamber to the tank boiler, although in some cases a comparatively large tube surface is arranged, so as to absorb the heat from the gases; yet it is possible that this extent of surface, encroaching on and causing the absence of sufficient space for combustion, may have such a cooling effect on the gases as to reduce them to a temperature below the igniting point, and so sometimes do harm. The most economical forms of these boilers have been where some attempt has been made to provide a space to serve the purpose of a combustion chamber.

Total heat of combustion of carbon and hydrogen.—The total amount of heat produced by the complete combustion of 1 lb. of carbon is found by experiment to be 14,500 thermal units, and this is sufficient to convert 15 lbs. of water at a temperature of 212° Fahr. into steam of the same temperature. This is spoken of shortly as the evaporation of 15 lbs. of water 'from and at 212° Fahr.' If the carbon be only imperfectly burned, so that carbonic oxide instead of carbonic acid is produced, the amount of heat generated is only 4,400 thermal units, which is less than one-third of the heat yielded by complete combustion. The evaporative power of hydrogen is 4·28 times as great as that of carbon, viz., about 62,000 thermal units per pound

Total heat of combustion of fuels generally.—The proper method of ascertaining the calorific value of actual fuels is by burning samples of them in the calorimeter, which gives results accurate and reliable, and, when exact knowledge is important, the instrument used is the bomb calorimeter of Messrs. Berthelot and Vieille. In this instrument, which is very simple, the powdered fuel is burned by pure oxygen

under pressure in an iron vessel under water. The ignition is caused electrically, and the rise of temperature of the water gives the heating value of the fuel. Thomson's calorimeter is also used for this purpose, but the necessary corrections for the latter instrument are more numerous than for the preceding.

Many formulæ have been devised in the endeavour to represent the total heat in terms of the chemical composition. The principal assumption made is that the value of the elements as existing in the fuel is the same as if they existed in the separate and uncombined condition, and that no heat is absorbed in the separation of the carbon and hydrogen of the hydro-carbons. It therefore ignores the heat of formation of the products and neglects entirely the latent heat. The following formula was originally given by Dulong, but it must be regarded as empirical and having no scientific basis, although it often represents the calorific value approximately, but gives generally somewhat too large a result, viz. :—

$$h=14{,}500\left\{C+4{\cdot}28\left(H-\frac{O}{8}\right)\right\}$$

where h=total heat of combustion of the fuel in thermal units, and C, H, and O are the fractional parts of carbon, hydrogen, and oxygen respectively contained in it.

For engineers, however, such formulæ have no value since the actual calorific value can be readily obtained, and much more easily than the chemical composition, and the latter, even when obtained, cannot be used to obtain the calorific value accurately.

It is sometimes convenient to be able to represent the total heat of evaporation approximately in terms of the chemical composition, but the formula should not be used indiscriminately for solids, liquids, or gases and as if it were scientifically correct, as is often done.

Evaporative power.—To convert 1 lb. of water at a temperature of 212° Fahr. into steam of the same temperature, 966 thermal units are required; therefore by dividing the total heat of combustion by 966, we get the number of pounds of water that each pound of fuel is theoretically capable of converting into steam from and at 212° Fahr. This is called *the evaporative power of the coal*, and on the basis of the above formula is represented approximately by

$$\frac{h}{966}=15\left\{C+4{\cdot}28\left(H-\frac{O}{8}\right)\right\}.$$

If, for example, we apply this formula to a bituminous coal, containing, say, 80 per cent. by weight of carbon, 5 per cent. of hydrogen, 5·5 per cent. of oxygen, and 9·5 per cent. of ash, &c., which latter has no evaporative power, we shall find that the total heat of combustion of 1 lb. of this coal is 14,268 thermal units, and this is theoretically capable of evaporating 14·77 lbs. of water. Again, if we take 1 lb. of Welsh steam coal, containing, say, 90 per cent. of carbon, 4 per cent of hydrogen, and 4 per cent. of oxygen, we find that it is theoretically capable of evaporating 15·75 lbs. of water. Therefore, if all the heat produced by the complete combustion of the coal could be utilised in the boiler, about 15 lbs. of water should generally be evaporated for each pound of coal burned. In practice, however, we fall far short of this. The best Welsh coal burnt in ordinary furnaces under the best

conditions only evaporates from 10 to 12 lbs. of water from and at 212° Fahr. per lb. of coal, and in general practice in marine boilers is less than this, and often considerably less.

Sources of waste.—The difference between the *available* evaporative power and its *theoretical* evaporative power is due mainly to the following causes :—

1. Waste of unburnt fuel in the solid state.
2. Waste of unburnt fuel in the smoky and gaseous states.
3. Loss of heat by the hot gas leaving by the funnel.
4. Waste by external radiation and conduction.

The waste of unburnt fuel in the solid state generally arises from brittleness in the fuel, and considerable air spaces between the fire-bars; the coal breaking into small pieces and falling between the bars into the ashpits. This loss may be reduced by firing evenly, disturbing the fires as little as possible, and taking care to burn all the small coal and cinders that may fall into the ashpits. With careless stoking it may be very considerable, but with the best firing and management it has been found to vary from 1 per cent. to 3 or 4 per cent.

The waste resulting from the escape of uncombined gases from the funnel can be prevented by providing a sufficient supply of air to the furnaces, admitted in a suitable place, and by good stoking. It is difficult to obtain perfect combustion of the gases with a moderate admission of air in any case, as the fires are so close to the tubes that the gases have to be distilled, mixed with air, and consumed in a short period and often in a confined space.

With care, the loss from this cause ought to be inappreciable. In six ships experimented on by the Institute of Mechanical Engineers, this loss, measured by the presence of carbonic oxide, was nothing in four ships, and 1·3 and 3·6 per cent. in two others. In two war ships recently the loss varied from nothing at low power to a maximum of 3 per cent. at high power. With improper stoking and bad arrangements for air admission it will be much greater.

Smoke.—The arrangements at furnace-doors and other parts for the supply of air above the fuel, for the gases, should always be used to assist in rendering the combustion complete.

It may, however, be a grave error to infer that when the *emission of smoke* has been prevented the desired end has been attained. Smoke forms from the quick cooling of the hydro-carbons temporarily dissociated by high temperature. Fuels containing no hydro-carbons, such as carbon and coke, do not give off smoke when burnt. If the combustion be complete, no smoke will be emitted, because the carbonic acid gas and steam formed are invisible. But carbonic oxide gas also is invisible, and if the combustion be imperfect and the products pass off in this form, no smoke will be visible, but the heat produced will be less than one-third of that yielded by complete combustion.

It should therefore be clearly understood that the absence of smoke is not necessarily a sign of economy. The great object to be attained is economy of fuel by rendering the combustion perfect, and the prevention of smoke will then follow as a natural consequence. It appears to be immaterial where the air is admitted, if it completely mixes with the gases before the latter are cooled below the temperature of ignition. It should not be admitted in volumes but in small jets, and this is done by causing the air to pass through perforated plates into the fur-

naces or combustion chamber, where it mixes with the hot gases, or, as in the Belleville boilers, by pumping jets of compressed air into the space immediately above the fire.

An efficient practical plan appears to be the fitting of suitable openings in the furnace door or frame, so that in the furnace of an ordinary cylindrical or water-tank boiler the air has to pass over the fuel. This allows more space and time for the mingling of the air with the gases, heating the former and preventing its cooling the gases below their igniting temperature, so that there is an increased chance of their complete combustion.

Mechanical stoking.—The maximum economy of coal can only be obtained by keeping a continuous supply on the fires and introducing a regular quantity of air for its combustion. The more continuous the supply of fuel the more certainly can it be properly consumed, but a continuous supply cannot be kept up while hand stoking is a necessity. The furnace door is wide open for some seconds during firing, allowing cold air to enter, and a large quantity of coal is thrown on the fire. The generation of gas is then large and sudden, and it is often not entirely burnt, whereas if the supply of fuel were regular the distillation of the gases would be gradual, and they would easily be burnt.

It would be a great improvement if mechanical stoking arrangements suitable for marine boilers could be designed; but the problem is a difficult one, and not any have been sufficiently satisfactory to warrant adoption, so that we are still dependent on the care and skill of the stoker, qualities frequently difficult to obtain. One of the advantages of the use of liquid fuel consists in the ease with which a regular and continuous supply can be provided. See Chapter VI.

Losses by hot gases leaving funnel—Funnel draught.—The third and principal cause of waste of heat is that due to the hot gases passing up the funnel. We will consider first the subject of draught. The draught of boilers with funnel only, is produced by the difference in weight between the hot gases in the funnel and that of an equal column of the external air, the adoption of any system of accelerated draught being equivalent to an increase in the length of the funnel.

Let T_1 = temperature of the air,
and T_2 = " " gases in the funnel.

It is easily proved that the weight of the gases discharged from the funnel per second is a maximum when

$$\frac{T_2 + 461}{T_1 + 461} = \frac{25}{12}$$

At the ordinary atmospheric temperatures this formula would give about 600° Fahr., or about that of melting lead, as the temperature of funnel gases that gives the most powerful funnel draught. If the temperature be increased beyond this amount, although the velocity of the gas in the funnel would increase, yet its volume would increase in a greater ratio, and consequently the weight of the gases discharged —that is, the draught—would be decreased.

The elevation of temperature of the products of combustion of coal of good quality, assuming the draught to be produced only by a funnel, so that double the quantity of air chemically necessary is

supplied, and assuming that the combustion is completed before any heat is abstracted, would be 2,400° to 2,500° Fahr., and as the maximum draught is produced with a temperature of about 600° Fahr. it appears that it is never necessary to expend more than one-fourth of the total heat of combustion, for the purpose of creating a draught by the funnel.

If the draught be accelerated and the arrangements such that one and a half times the air chemically necessary is sufficient, the elevation of temperature on the same assumption would be about 3,200° Fahr., while in this case no elevation of temperature above that of the external air is necessary for the gases in the chimney. We see, therefore, that with accelerated draught or forced combustion the boilers are for a double reason capable of greater economy than if with funnel draught only.

With funnel draught only, if the funnel is large enough in area to give sufficient draught with a less funnel temperature than 600° Fahr., the temperature can be reduced economically below this point by additional heating surface, or by the use of retarders in the boiler tubes, but it is never advantageous to increase it. With accelerated draught the only limit to the reduction of funnel temperature, and waste from this cause, is that imposed by the extra weight and cost of the additional heating surface required, whether this take the form of additional boiler-heating surface, feed-water heaters in the uptakes as in the Belleville economiser, or air-heating appliances in the uptakes, as in Howden's, and Ellis & Eaves' systems.

A portion of this waste heat used to be extracted by the old superheaters,[1] but these were not efficient as heat abstractors, owing to the low specific heat and bad conducting powers of the steam contained in them. The feed-water heaters or economisers are much more efficient as heat abstractors. The heat thus abstracted from the funnel gases is a clear gain, and reduces the amount of this loss, which is always the principal one in boilers.

The loss of heat from this cause increases with the rate of combustion. At the higher rates it is often from 25 to 30 per cent., but much less at lower rates of combustion, when the proportion of heating surface to coal burnt becomes increased, while with extreme forcing the loss may be very much greater than this figure.

Temperatures of gases in the boilers.—It will be interesting to see the distribution of temperature in the interior of a boiler when at work. Some experiments on this point were made recently at Devonport on a marine return tube boiler worked on shore. The boiler had two furnaces and 2¾-inch tubes, 6 ft. 8 in. long, and was burning coal at the rate of 17 lbs. per square foot of grate with funnel draught only. The temperatures were taken by a Le Chatelier thermo-electric pyrometer.

Under these circumstances we saw above that if double the air chemically necessary be admitted, the elevation of temperature of the products, assuming combustion to be completed before any heat is abstracted, would be 2,400° Fahr. This condition is, however, never realised in practice, since the abstraction of heat goes on in the furnace continuously, and the combustion is not entirely completed

[1] See Chapter XIII.

there. The temperature of the gases in the furnace will, therefore, be less than 2,400° Fahr. The experiments showed that in the combustion chamber the temperature was 1,576° Fahr.; in the tubes it varied from 1,302° Fahr. just inside at the combustion chamber end, to 795° Fahr. near the exit; the temperature in the smoke-box being 740° Fahr. Fig. 15 shows this graphically.

With severe forcing, as in torpedo boats and destroyers, the funnel temperature is much higher; 1,444° Fahr. has recently been recorded.

In the 'Powerful's' Belleville boilers, on shore, with clean tubes and fresh water, and with careful stoking so that combustion all occurs below the tubes, conditions favourable to efficiency of heating surface, and burning 24 lbs. of coal per square foot of grate, it was about 650° Fahr. by pyrometer. With the newer Belleville boiler, with 'economiser' in the uptakes, the temperature on exit from the tubes is only

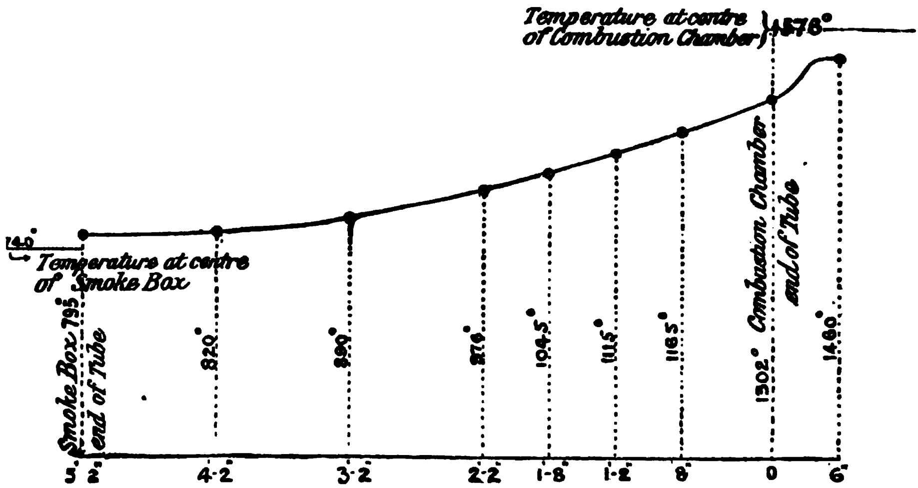


FIG. 15.

about 500° Fahr. with the same rate of combustion and favourable conditions of trial on shore. Under sea-going conditions, however, the heating surface is less clean and efficient, while the stoking is not so good, so that some combustion probably occurs among the tubes, and these latter temperatures are exceeded.

Radiation and conduction.—The loss by radiation and conduction is taken as the remainder after the three preceding losses have been measured. The loss by radiation depends largely on the size of the boilers for the power being exerted, or rather the extent of surface for radiation per pound of coal burnt. This will be smaller, in the same type of boiler, the smaller is the size of the boilers for the power developed. Although the whole of the surfaces that can be so treated are covered with a lagging of silicate cotton, this serves only to reduce the waste, which is always appreciable and sometimes considerable. With boilers of ample size for power developed, and with high steam pressure, the loss may rise above 10 per cent.; but such arrangements could not be regarded as satisfactory. With well-designed arrangements,

well clothed with non-conducting material, the loss from this cause, or the loss not accounted for by either of the three preceding causes, should not exceed 6 per cent.

Efficiency of marine boilers.—The ratio borne by the heat actually transmitted to the water in the boiler, to the total heat developed by the combustion of the fuel, is called the *efficiency of the boiler*.

The several causes of waste enumerated above are always at work, so that the total heat that should be yielded by the complete combustion of the coal, is not available for transmission to the water. The total waste depends greatly on the skill exercised in the design of the boiler and the care displayed in its management. In a well designed marine boiler, with careful firing, the waste amounts to 30 per cent. of the total heat of combustion of the coal. If the management be careless or unskilful the loss will considerably exceed this.

The efficiency of boilers is determined at various rates of combustion by trials, usually carried out on shore, during which the following data are accurately recorded at regular intervals: (*a*) Quantity of feed-water admitted to the boiler; (*b*) quantity of coal burnt; (*c*) steam pressure in the boiler; (*d*) temperature of feed-water. The steam generated on shore trials usually escapes through the stop valve into the atmosphere. Calculations, as in the succeeding example, based on such records, determine boiler efficiency with sufficient accuracy for practical purposes, but the following assumptions are made, which would require verification in more elaborate experiments.

It is assumed on such trials: (*a*) that the steam produced is dry and saturated; (*b*) that all water fed into the boiler is evaporated, and leaves the boiler as steam; i.e. no losses occur by water leakage through glands, joints, etc.; (*c*) that the quantity of water contained in the boiler, and the condition of the fires, is the same at the end as at the beginning of the trial. Of these assumptions, the first would be rendered unnecessary if the steam generated were collected, and its degree of dryness determined by a calorimeter, instead of being permitted to escape into the atmosphere; the validity of the second could be verified either by condensing and weighing the steam generated, or by water-testing the boiler and fittings after trial; and any errors arising from the last assumption would be negligible if the trial were of sufficiently long duration, and the feeding and stoking carefully managed especially towards the end of the trial.

In the mercantile marine cylindrical boilers are generally fitted, and are of ample proportions for the power required of them. They are usually provided with special devices, such as air-heating appliances, etc., which tend to increase the boiler efficiency. In a trial carried out under ordinary conditions at sea under the supervision of the Admiralty Boiler Committee with the cylindrical boilers of s.s. 'Saxonia,' the efficiency of the boilers reached the high value of 82·3 per cent., burning coal at a rate of 20 pounds per hour per square foot of grate, at which rate approximately full power is obtained.

The limitations of weight and space imposed upon installations of naval boilers necessitate the use of boilers of comparatively small size for the power required, and generally involve some sacrifice of economy as compared with boilers fitted in vessels of the mercantile marine where such limitations do not apply with equal force. Average values

of the boiler efficiency for boilers used in H.M. Navy, as deduced from shore trials and from trials at sea, are given in the following table :—

Type of Boiler	Value of Boiler Efficiency per cent.		
	At about $\frac{1}{2}$ Power	At about $\frac{3}{4}$ Power	At about Full Power
Babcock & Wilcox . .	78	75	70
Belleville	77	76	70
Cylindrical	70	64	63
Dürr	75	73	70
Niclausse	73	70	66
Yarrow (large tube) . .	73	67	67

The water-tube boilers in the above table are of the large tube type. For torpedo craft, where it is imperative to reduce weights as much as possible, water-tube boilers of the small tube type are fitted ; the economy of these boilers is less than that of the large tube type, this relative want of economy being part of the price paid for the advantage of reduced weight. From a number of trials with water-tube boilers of the small tube type, the boiler efficiency at full power rarely exceeds 58 per cent., and, at about half-power, 64 per cent.

The efficiency of marine boilers of the better class, therefore, varies between 60 to 80 per cent. Below 60 per cent. it is considered bad.

Examples of efficiencies.—The following example from actual practice will indicate the method of calculating the boiler efficiency when the trial data are known ; in this case the records were obtained on evaporation trials on shore :—Coal burnt per hour = 864 pounds ; feed entering boiler per hour = 8,350 pounds ; temperature of feed-water = 100° F.

The steam was blown off into the atmosphere at 275 lbs. per square inch. From these records the boiler efficiency is calculated as follows, the calorific value of the coal (determined subsequently by calorimeter) being 15,120 British thermal units.

The total heat per pound of dry saturated steam at 275 lbs. per square inch (gauge) calculated from feed-water at 32° F. is 1208·3 British thermal units, while the heat added per pound of feed-water leaving the boiler as dry steam at 275 lbs. per square inch is $\{1208{\cdot}3 - (100 - 32)\} = 1140{\cdot}3$ British thermal units ; hence the heat carried away by 8,350 pounds of steam is $1140{\cdot}3 \times 8350 = 9521505$ British thermal units, and as the heat of combustion of 864 lbs. of coal is $864 \times 15120 = 13063680$ British thermal units, therefore boiler

$$\text{efficiency} = \frac{9521505}{13063680} = {\cdot}73 \text{ nearly.}$$

The approximate distribution of heat, under sea-going conditions, in a fair example of cylindrical marine boiler, supplied with a sufficiency of air and burning about 20 lbs. of coal per square foot of grate, the heating surface being about thirty times the grate, would be as follows :—

	Per cent.
Absorbed by feed water	68
Wasted in funnel gases	24
Waste by unburnt carbon in ashes	2
Waste by imperfect combustion	0
Balance accounted for by radiation, &c.	6
	100

The first number, 68 per cent., represents the efficiency of the boiler.

CHAPTER V.

METHODS OF ACCELERATING THE RATE OF COMBUSTION OF FUEL.

Accelerated draught generally.—The question of weight and space occupied by the machinery on board ship is one of great importance. With natural draught only, the rate of combustion per square foot of fire-grate in marine boilers is comparatively slow, even under the most favourable circumstances, as the height of the funnel is necessarily limited, and if the natural draught were alone depended on, the boilers would in many cases require more space than could be allotted to them, especially in warships. It is therefore necessary that methods should be adopted to increase the rate of combustion in the fires, and consequently the generative powers of the boilers, in order to obtain the required power in a limited space.

Steam blast.—An old plan adopted for forcing the draught in marine boilers was by admitting a jet of steam from the boilers at the base of the funnel; this is usually known as the 'steam blast.' It can be very readily applied, and the rate of combustion can thus be increased by about 25 per cent.; but it is a very extravagant way of obtaining increased power, and in these days of high-pressure steam, quite inadmissible, owing to the loss of fresh water involved. As an example of the expense of this means of producing draught, it may be mentioned that when burning 30 lbs. of coal per square foot of grate by the aid of the steam blast, the steam used by the blast is 10 per cent. of the total steam produced.

A modified form of this plan is in general daily use in the railway locomotives, but it is peculiarly adapted to the conditions under which they have to work. In these no attempt is made to condense the steam after its utilisation in the cylinders, and it is exhausted at a comparatively high pressure from them to the funnel direct, thus causing the blast; special arrangements are made to provide the necessary supply of fresh feed-water. Hence the steam used for blast purposes would otherwise do no useful work, and in this way a simple, and under the circumstances efficient, means of accelerating the draught is obtained.

Principal plans.—The principal other plans tried are:—

1. Creating draught by admission of steam in a closed ashpit.
2. Admitting jets of compressed air into the base of the funnel in a similar manner to the steam jet.
3. Fitting a centrifugal fan in the uptake to draw off the products of combustion.
4. Blowing the air into closed ashpits.
5. Closing the stokehold and keeping it filled with compressed air.

Air jets in funnel.—The admission of compressed air to the base of the chimney was tried to some extent in France, but when the success of torpedo boats fitted with closed stokeholds was demonstrated, competitive trials were made in that country with various systems, viz. with air jets in the funnel, using exhausting fans in the uptakes, and using fans in a closed stokehold. The expense of production of draught was found to be in favour of the closed stokeholds, and the latter were then definitely adopted in the large vessels, although it does not appear that the evaporative power of the fuel with the three systems was ascertained, it being assumed apparently that the effect of a certain draught—whether produced by stokehold pressure, or air, or fan suction in the funnel—was the same. A small installation has recently (1895) been fitted in the 'Wild Swan,' to assist in maintaining the power under adverse conditions of draught, but reports received show that no benefit is derived from it, while the consumption is sensibly increased. This system has, therefore, had practically no application of importance.

Fans in uptake or induced draught.—The fitting of a centrifugal exhaust fan in the funnel, through which the whole of the gases from the boiler must pass, has been tried in the Royal Navy in the 'Gossamer' with locomotive boilers, and many experiments were carried out in that vessel. It has since been applied to the battleships 'Magnificent' and 'Illustrious' and gunboat 'Torch.' The arrangement consists essentially of a single fan in the uptake of each boiler of considerably larger dimensions than required with closed stokeholds. A second gunboat, the 'Alert,' with 'closed stokehold' fittings, but in other respects exactly similar to the 'Torch,' was built at the same time, and this has enabled comparative tests of long duration to be carried out. These tests have shown no economy in favour of induced draught as has been claimed; while the fans and fittings sometimes become overheated to an extent which causes these appliances to fail. The advantage of the open stokehold, however, remains.

Closed ashpits.—The fourth plan, viz. blowing air into closed ashpits, is an efficient method of increasing the power of boilers; but it necessitates closed ashpits, as the pressure in the furnaces is greater than that in the stokehold, and unless proper precautions be taken before opening the furnace doors at the time of firing, the flame may be blown into the stokehold, with possibly dangerous consequences.

This plan is often adopted in small steamboats in the Royal Navy, as it is a simple means of increasing the draught. It has also been fitted in many vessels of the American and other navies, and many ships of the mercantile marine. As the air for combustion does not pass through the stokeholds, special arrangements are provided to supply air for the men engaged there. This system of forced draught is generally preferred in vessels of the mercantile marine, and the arrangements are generally such that the air supply to the ashpits is automatically closed by the operation of opening the furnace door. As the pressure over the fires is rather greater than that in the stokehold, it is necessary, in order to secure admission of air through the fire-doors, that the latter should be surrounded by a chamber in connection with the ashpit, so that the required pressure of air will be obtained.

Closed stokeholds.—The fifth plan, by which the stokeholds are made

airtight and filled with slightly compressed air by means of blowing fans, has been generally fitted in vessels of the Royal Navy where forced draught has been required. This system was first adopted by the designers of torpedo boats, and very high powers were obtained from their boilers when worked under air pressure. In these original boats only one boiler was fitted, so that the application of the system was more simple than in the case of vessels containing a number of boilers.

The latest method of applying this system in the Royal Navy is shown in Figs. 77 and 78. The stokeholds are enclosed for the purpose of being placed under air pressure, by fitting vertical screen plates carried down between and at the ends of the boilers to meet the front boiler bearers. These screen plates are worked around the fronts of the boilers to enclose the smoke-boxes, so as to keep the stokeholds cool, and are carried back sufficiently far at the sides of the boilers to clear the water gauges. In this figure the top of the enclosure is formed by the steel, or protective deck. In many cases, especially of the older vessels, where there is a considerable space between the tops of boilers and the deck, the top of the airtight enclosure is formed by fitting a horizontal ceiling, extending from the coal bunker bulkhead to the front of the boiler, at about the level of the top of the boiler, as illustrated in Figs. 19 and 20. The stokeholds are thus made into closed airtight chambers of comparatively small dimensions. The screens are shown in thick lines.

Débris deck.—The airtight ceilings of the stokeholds, when fitted separately, usually form portions of the 'débris' or 'splinter' deck generally fitted over the openings in the machinery department to protect the steam pipes and fittings from injury from fragments of shot, shell, or other débris. This débris deck then serves for carrying the fans for producing air pressure in the stokeholds.

Air-locks.—In order to provide for passage to and from the stokeholds when under pressure, air-locks are fitted. These consist of small airtight chambers fitted with two-hinged doors opening against the air pressure, as shown in Fig. 78, and also in Figs. 19 and 20. In passing through, one door only is open at a time, which makes it possible to enter or leave the stokehold without allowing much air to escape, and so reduce the pressure in the stokehold. Air-locks are necessary at all places at which communication is made between the compartments under pressure and any other part of the ship.

In the stokeholds of most of the fast cruisers no special horizontal ceilings are required to be fitted, as the deck of the ship answers the purpose. All that is necessary is to carry vertical screen plates around the boilers from the deck to the boiler bearers, so as to isolate the stokeholds. The other fittings are similar to those in the armour-clad ships, modified in detail as required to suit the different arrangements of the ships.

Advantages of closed stokeholds.—Forced draught in general possesses the important advantage for warships that a great reduction can be made in the space and weight required for the boilers; the extra power necessary for full-speed working, instead of being obtained by the provision of additional boilers, which would occupy much space and weight, although but seldom required, is provided by

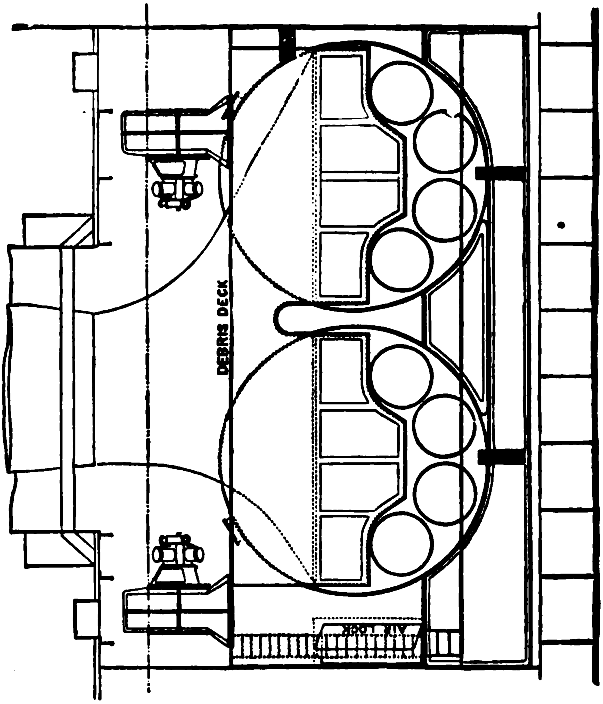


FIG. 20.

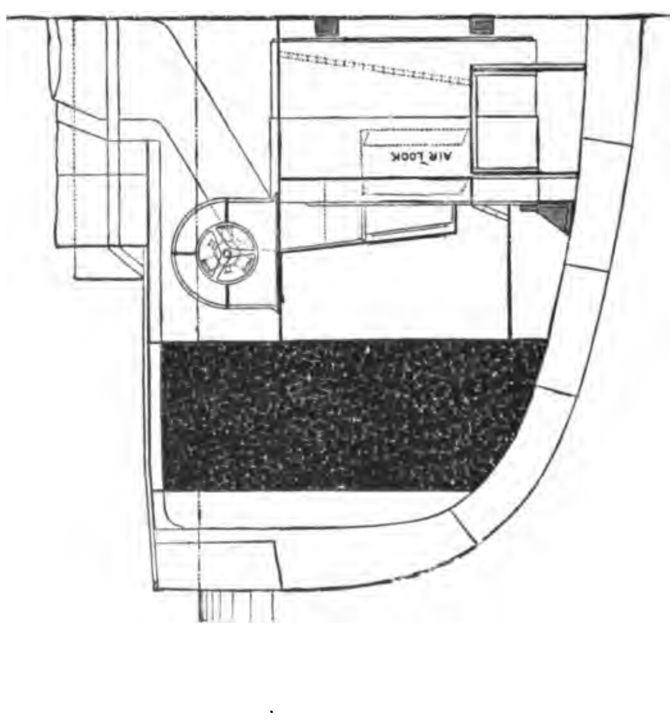


FIG. 19.

the addition of fans and screens, which are comparatively inexpensive and involve very little additional space and weight.

The openings in the deck for the boiler rooms may be reduced to the minimum required for the supply of air to the fans, for the funnels, and for convenient access to the boiler rooms. The supply of air to the boiler rooms being entirely provided by the fans, the power of the ship is practically independent of the wind, which is a matter of importance, especially in the Tropics, and the power developed can be easily regulated by the speed at which the fans are driven.

The first ships in the Royal Navy to which this system was applied were the sloop 'Satellite' and the turret ship 'Conqueror' in 1882. During the four hours' full-power trial of the 'Satellite' with natural draught, 10·15 I.H.P. were developed per square foot of fire-grate. With an air pressure in the stokeholds equal to 1½ to 2 inches of water, 16·9 I.H.P. per square foot of firegrate were obtained, being an increase of 66·5 per cent. In the 'Conqueror,' also, the gain in power with a mean air pressure of 1¾ inches over that obtained with natural draught was 68·6 per cent.

Large numbers of vessels have since been fitted with boilers on this system, and where not carried to extreme limits it has given satisfaction. As an example of the results obtained, that of the 'Sanspareil' may be mentioned. This vessel was tried in 1888, and with a grate surface of 722 square feet and total heating surface of 19,980 square feet, developed 14,483 I.H.P. for four hours with 2 inches of air pressure, or 20 I.H.P. per square foot of grate surface.

Since this period, however, experience has shown the desirability of reducing the amount to which the boilers are forced, and the last Admiralty specifications for water-tank boilers provide a total heating surface of not less than 2·5 square feet per I.H.P. at natural draught power, and 12 to 12½ I.H.P. per square foot of grate, while the forced draught power is limited to 20 per cent. beyond the natural draught power.

Air heating systems.—It was seen on page 40 that the amount of heat passing up the funnel and wasted is very considerable, and various plans are in operation to reduce it. The most extensively used consists in heating the air passing to the furnaces for combustion, by the escaping hot gases. This is effected by fitting in the uptakes of the boilers a series of thin tubes, through or around which the air for combustion is made to pass, and on the other side of which are the hot gases on their way to the funnel.

The combination of this system with the closed ashpit method of producing the draught and other modifications of detail, is known as Howden's system. The combination of air heating with the induced draught caused by fitting fans in the uptake, produces Ellis & Eaves' system, introduced and developed by Messrs. Brown, of Sheffield.

Howden's system.—The development of the air-heating principle in this country is due principally to Mr. Howden, of Glasgow. The air-heating appliances in his system consist of a considerable number of thin vertical tubes arranged in a chamber immediately over the smoke-box, and through which tubes the escaping hot gases pass. The air for combustion is delivered by the fans through a pipe, enters this chamber at the middle, and proceeds past these tubes on either side,

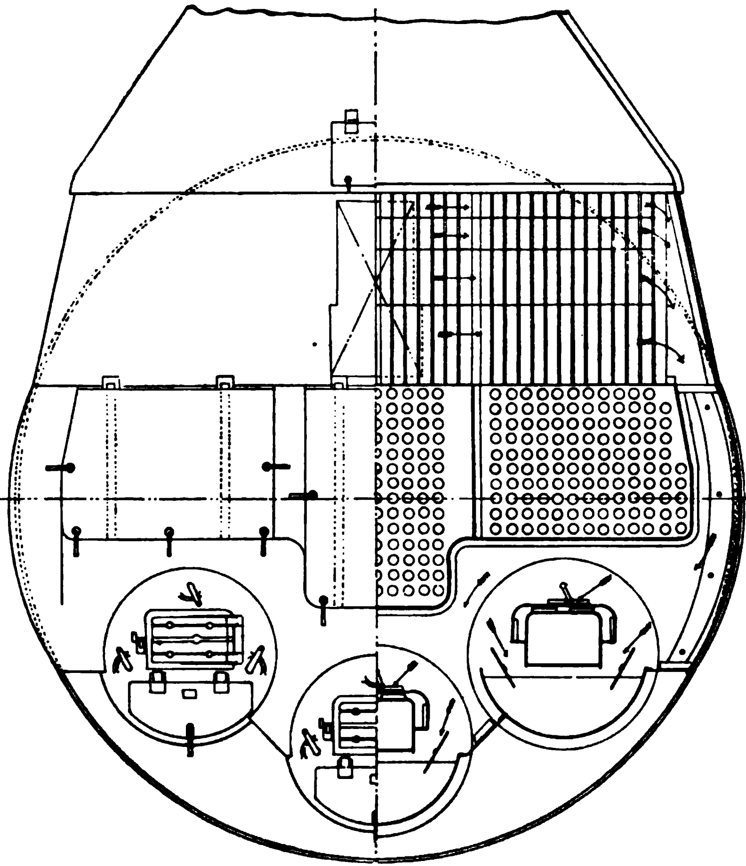


FIG. 21.

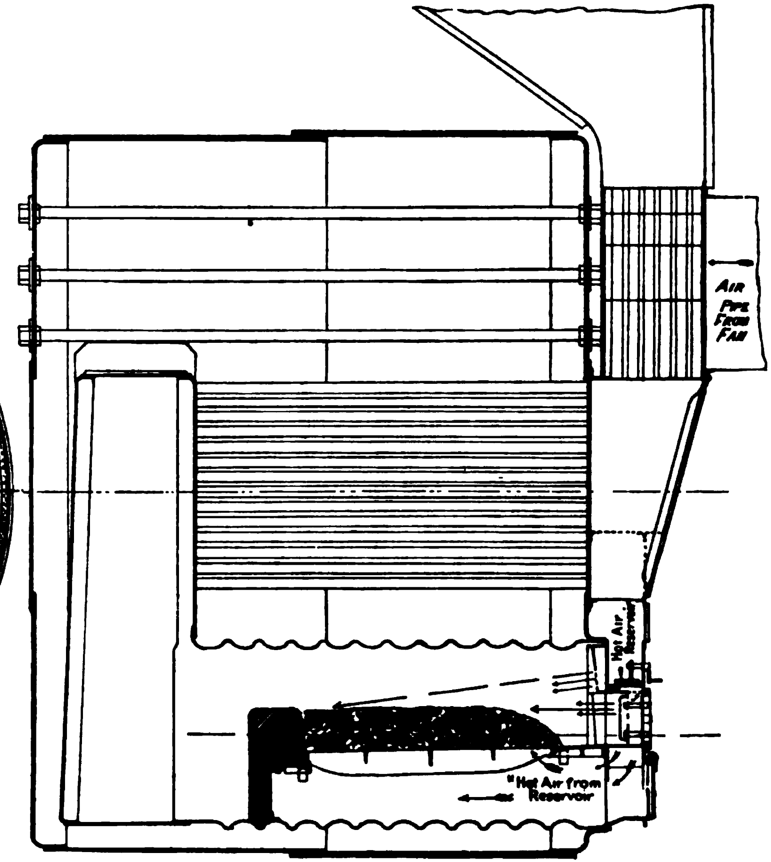


FIG. 22.

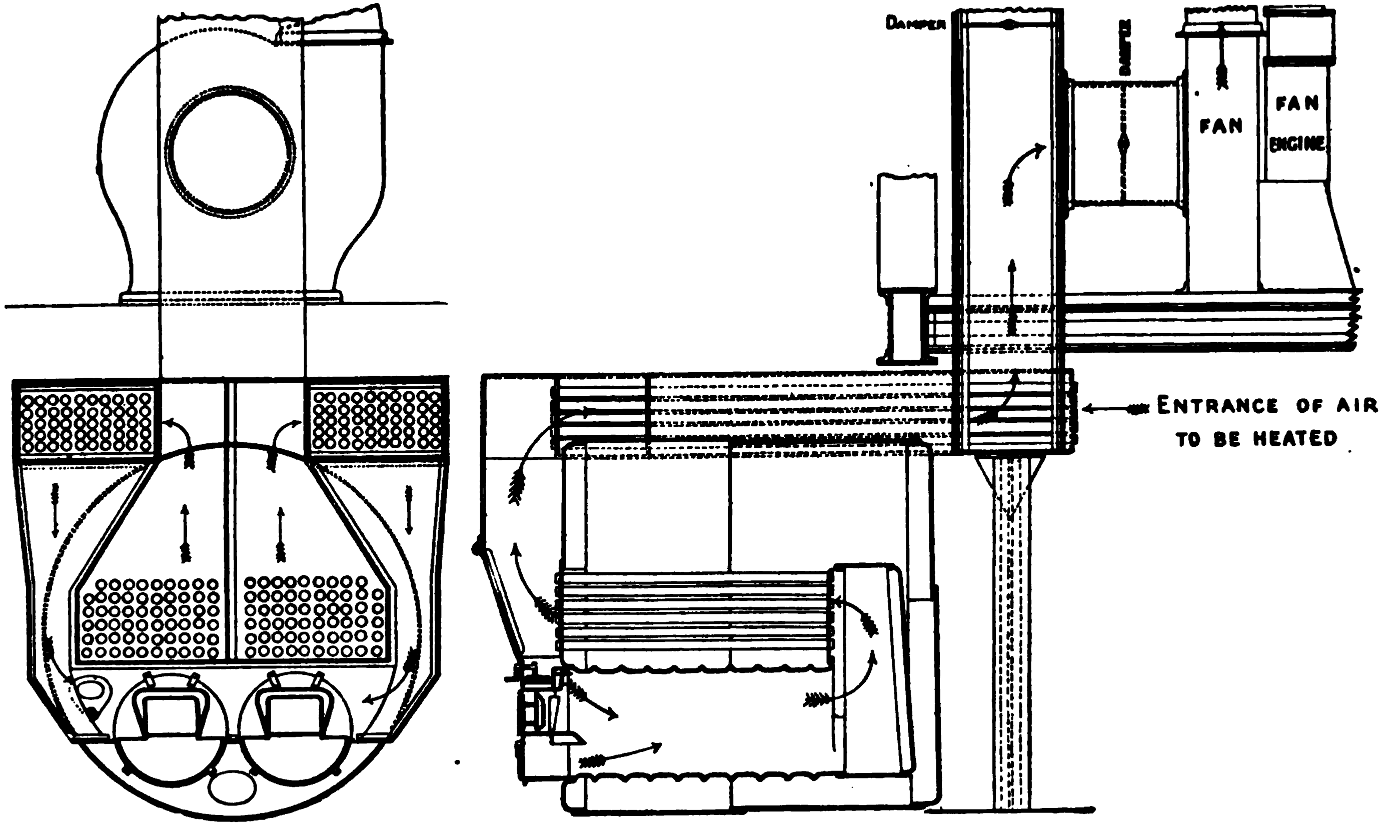


FIG. 28.

gradually increasing in temperature by contact with them, and being delivered through channels formed on either side of the smoke-box to closed spaces in connection with the ashpit, and also in the furnace frame. Valves are fitted to admit a certain proportion of this heated air under pressure to the ashpit, and also a certain proportion to the furnaces over the fires, the proportions being ascertained from experience. The front of the ashpit is closed as in the closed ashpit system. The details of the apparatus are shown in Figs. 21 and 22. Special strips of twisted plate are also placed in each boiler tube, called 'retarders,' which cause the gases traversing these tubes to take a spiral course and be retained in contact with the boiler tubes for a longer period than they otherwise would. Mr. Howden's grate is a very short one, so that the ratio of heating surface to fire-grate is considerably more than usual. Howden's system is fitted in a very large number of vessels of the mercantile marine, and also in several ships of the American Navy, satisfactory results being obtained.

The closed ash-pit system of draught in association with heated air for combustion is fitted in some large cruisers in the British navy both for cylindrical boilers and also in connection with water-tube boilers of the Yarrow type.

Ellis & Eaves' system.—This combination of draught by exhaust fans in the uptakes with air-heating appliances was introduced at a later period than the Howden system. In this plan the air-heating tubes are arranged in nests of horizontal tubes of considerable length on top of the boiler. Unlike Mr. Howden's arrangement, the escaping hot gases are passed along on the outsides of the tubes on their way to the fan in the uptake, while the air for combustion enters the ends of the tubes from the open stokehold and passes through them (see Fig. 23). The heated air is conducted, as before, by channels at the sides of the smoke-boxes to the ashpits, and also to spaces around the furnace frames. A certain amount of air is admitted above the fires as well as through the ashpits. The front of the ashpit is closed, as in the last system mentioned. This plan has been fitted to several vessels of the mercantile marine, and satisfactory results have been reported.

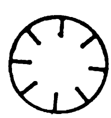


FIG. 23a.

With the fittings as arranged by Messrs. Brown, and also in large numbers of boilers not working on this system, a special form of boiler tube is used, known as the 'Serve' tube (Fig. 23a). It consists of an ordinary tube with the addition of several internal projecting ribs, which conduct an additional quantity of heat from the escaping gases to the surrounding water. These tubes add to the efficiency of a boiler, but their weight is about double that of ordinary plain tubes.

CHAPTER VI.

PETROLEUM AS FUEL.

Recent progress.—A few years ago a great advance was made in the use of petroleum as fuel for generating steam in marine and locomotive boilers, the increased production of cheap crude oil from the many new oilfields in the United States having extended its use for fuel wherever its cost compared favourably with that of coal.

The reduction in the cost of transport by the adoption of pipe lines for its conveyance has been the chief agency in rapidly promoting its use, for both the east and west seaboards of America have now pipe lines for fuel oil leading from the sources of production to ports of shipment, the most recent conveying thick oil a distance of nearly 300 miles to San Francisco Bay through an 8-inch pipe with pumping stations every twenty miles.

The oil-fields of Roumania and Burmah are now established sources of oil fuel, and other sources are being actively sought for within and without British territory.

Description of liquid fuel.—The term liquid fuel may be conveniently applied to any hydro-carbon compound which either in its natural or prepared state may be sufficiently fluid to flow by gravity or pressure from one position to another. The term is equally applicable to natural mineral oils, the refuse tarry products from gasworks and blast furnaces, the oils obtained from the distillation of shale, and all vegetable and animal oils, and it would also properly include pitch, bitumen, and wax, as it would be possible to effect the combustion of each of these hydro-carbons with the appliances in use for any of the varieties of oil.

Natural mineral oil, as raised direct from the oil-bearing strata, is known as crude oil, and, in addition to water, earthy impurities, and gas with which it may be charged, is found to consist of a number of hydro-carbons chemically combined in slightly different combining proportions, and as each of these different combinations has a different

vapourising temperature, some of which are very low, the crude oil is subjected to various distillations which separate out the several commercial products of heavy and light oils, including naphtha, gasolene, benzine, lubricating oils &c., and it is only the residue remaining which is suitable for service boiler fuel with safety.

Flash point.—Even these residues vary considerably in their combinations of hydrogen and carbon, and their relative freedom from easy inflammability varies as indicated by their 'flash point' temperature, i.e. the temperature at which the gases given off can be ignited by a flame when mixed with air in a special testing apparatus. At present, for the avoidance of any possible risk the flash point temperature for British naval purposes has been limited to 200° F. as a minimum, but it should be remembered that gases are given off from it at a temperature of even 50° F. below the flash point.

Chemical composition.—The mean average chemical composition of the residues now used for fuel, may be taken as 84 per cent. of carbon, 13 per cent. of hydrogen, with 2·5 per cent. of oxygen, and 0·5 per cent. of sulphur, and the total heat of combustion of one pound of this fuel calculated from this analysis would be in round numbers 20,000 British thermal units, or at least one-third more than given by the best Welsh coal. The specific gravity is about ·9.

Early use.—The use of petroleum for boiler fuel originated at the first great source of supply, *i.e.* on the shores of the Caspian Sea, and both in marine and locomotive boilers in that region, the earlier attempts to burn the thick liquid was to introduce it into the furnaces in shallow pans, into which it flowed by gravity, and many devices were resorted to for assisting its combustion by slowly passing it from one pan to another after successive heatings, the air supply during the operation traversing the furnace as in the case of coal burning. This method of burning it was soon found to be quite inadequate to meet the requirements for steam beyond a very slow rate, for whatever quantity of oil was introduced into the furnace, combustion could only be effected over the limited surface of the liquid by the actual contact of the hydro-carbon molecules forming that surface, with oxygen at a sufficiently high temperature to admit of combustion. This constitutes the chief mechanical difficulty in providing for the combustion of liquid fuel at rates at all similar to that of a solid fuel, such as coal.

Comparison between the burning of oil and coal.—As each molecule of fuel must be brought into contact with its combining proportion of oxygen before combustion is effected, in coal-burning furnaces this is mechanically provided for by spreading the coal on a firegrate surface, through and over the whole area of which when once the fire is alight, air enters at a rate depending on either chimney or forced draught. The whole mass is then in a condition favourable to offering an extremely large surface to the entering air which is compelled to traverse through it before proceeding to the funnel, and is thus brought into contact with a very large number of fuel molecules.

With liquid fuel, however, it is not possible to provide for the passage of air for combustion through the mass in a similar manner; but it is possible to break up the mass on its entrance into

the furnace by projecting it through an instrument which discharges the liquid in a finely divided spray, and in devising such instruments much ingenuity has been expended.

Liquid fuel burners.—These instruments were originally known by the name of 'sprinklers'; but the general term 'burner' is now usually applied to all the varieties. They may be conveniently divided into two classes: (1) 'Pressure burners,' which effect the desired pulverisation by discharging the liquid through a very small orifice under considerable pressure; (2) 'Steam or air burners,' which break up the oil by an elastic fluid such as steam or air, the issuing jets combining with the stream of oil either inside the instrument, or at the point of discharge. This class also includes those which use both steam and air, the latter entering the instrument at normal pressure, and joining the issuing steam and oil by the inductive action of the jet.

There are but few varieties of the first class, one being simply a nozzle at the termination of the oil supply pipe with a small hole of from one-sixteenth to one-sixtieth of an inch in diameter. Examples of this form have been used by Mr. Howden in combination with his system of heated air forced draught for boilers, and one is described below.

Application to a cylindrical boiler.—Each furnace front has an air casing supplied with air from a fan delivering it to the furnace through heating tubes in the uptake. Two burners are fitted over each furnace door, the prolonged stems of the burners projecting through the air spacing, and each discharging in the centre of a circular opening through which air is admitted which accompanies the oil streams into the furnace. The burners are practically pipe branches from the main supply, each with a suitable valve for controlling discharge, which takes place through an orifice about one-sixteenth of an inch in diameter. The oil in the supply pipe is maintained at a constant pressure which does not generally exceed 200 lbs. per square inch, but may be as low as 25 lbs., according to the rate of discharge required. The oil is circulated through a heater, where its temperature can be raised to about 200° F. Filters are fitted in the oil discharge pipe in order to extract any solid matter which the oil may contain, and so prevent it from choking the sprayers. The fire bars are retained in place and covered with fire tiles, and in the middle of the furnace a transverse bridge is built, leaving about one-third of the sectional area of the furnace for the flames and products of combustion to pass over into the combustion chamber beyond.

Korting sprayer.—Another example of pressure burners with a more extended application is known as the 'Korting' centrifugal sprayer, the nozzle being arranged with a coarse helical channel immediately inside the point of emission, and it is claimed that the passage of oil under pressure through this channel imparts to it a centrifugal force which helps its projection as a cone of fine spray. This arrangement also requires the oil to be maintained under constant pressure and at a temperature of 200° to 250° F. An example of this sprayer is shown in Fig. 24.

Other types of similar instruments rely upon some special shape

of orifice to finely spray the oil, but whatever the type, high pressure is necessary, and also high temperature for all but the thinner oils.

Steam or air sprayers.—The second class of instruments are very numerous, but the underlying principles are the same.

Most of the varieties adopt concentric orifices in the instrument, through one of which the oil flows either by gravity or pump pressure, the other discharges the pulverising steam or air also under pressure. Where the concentric form is not resorted to, the issuing jets of oil and steam or air are brought together at a common point of discharge, the spraying agent being directed across or oblique to the petroleum stream as in a blow-pipe. Burners which also admit in the same instrument a supply of air for combustion, do this through a third opening, leading to another annular space for discharge near the oil stream.

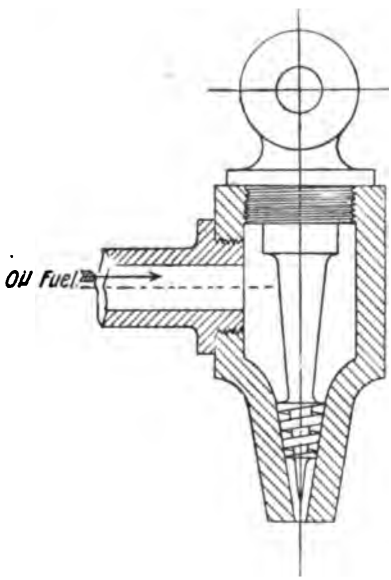


FIG. 24.—KORTING'S BURNER.

The best known forms of steam spraying instruments in use in Great Britain are the 'Holden,' the 'Rusden and Eeles,' and the 'Orde' burners. A type known as the 'Kermode' burner, using compressed air as the spraying medium, and arranged with helical channels to impart centrifugal motion to the oil and air at the point of discharge, has recently been worked with good results.

Object sought.—The chief object of each particular device is to project the oil into the furnace in the finest state of division as spray, and the best forms accomplish this by heating the oil to a temperature approaching that of its flash point, some even attempting to convert it into vapour before emission, but with doubtful success where considerable quantities have to be dealt with in limited time and space. The common practice with steam sprayers is to superheat the steam, and with air sprayers to heat the air, by passing the supply pipe through some convenient part of the furnace or uptake.

With respect to the atomising of the fuel, the best types of modern burners may be regarded as effecting this equally well; some having advantages in controllability, some in simplicity.

Means are fitted for regulating the amount of petroleum and steam supplied to suit various rates of combustion, but the adjustment requires considerable care and attention, so that it is often recommended to design the proportions of the nozzles for a given rate of burning, and to provide for increase of power by additional nozzles, which may be started or shut off. By this plan the range of adjustment of each nozzle is either much limited or absent.

Holden's burner.—Mr. Holden, of the Great Eastern Railway, was successful in applying liquid fuel to many locomotives and other boilers, and his burner may be taken as a typical example of steam spraying. In his arrangement the air is admitted through a central pipe, steam is supplied through an annular space outside it, while the oil to be burned is admitted to another annular space outside the latter. The speciality in Holden's burner, shown in Figs. 25 and 25*a*, lies in the fact that the steam is not only admitted inside the burner to spray the oil, but is also admitted to a pipe B bent in the form of a ring A near the mouth of the burner, with small holes from which the steam rushes, and is directed on the issuing jet of petroleum and steam,

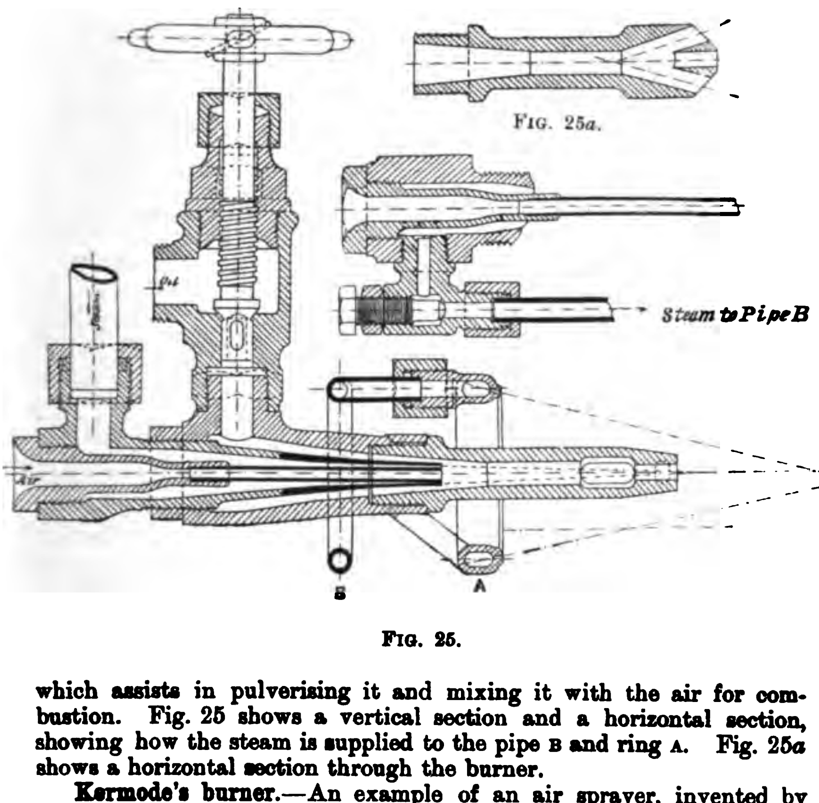


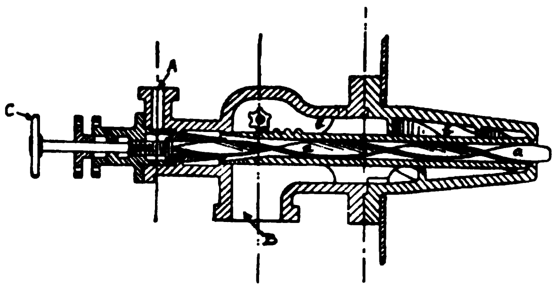
FIG. 25.

which assists in pulverising it and mixing it with the air for combustion. Fig. 25 shows a vertical section and a horizontal section, showing how the steam is supplied to the pipe B and ring A. Fig. 25*a* shows a horizontal section through the burner.

Kermode's burner.—An example of an air sprayer, invented by Mr. Kermode, is shown in Figs. 26 and 26*a*. In this case the oil is supplied through the passage A, and, in its course through the central channel *a a* acquires a rotary motion by means of the helical form of the spindle. Compressed air is admitted through the channel B, which also acquires a rotary motion by means of the helical vanes in the annular space *b b*. The helices are arranged so that the rotary motions of the air and oil are in opposite directions, and the two meet and mix together as they emerge from the nozzle. The rate of

supply of oil is regulated by the wheel and screw C, and that of the air by the wheel and rack D.

Air necessary for combustion of oil fuel.—In order to effect complete combustion of oil fuel after its projection into a furnace it will be necessary to ensure the provision of sufficient oxygen, and as air is the vehicle for its supply, the least quantity which will furnish the necessary oxygen for complete combustion of oil fuel will be found, from the chemical composition previously given and the rules in Chapter IV., to be 14 lbs. of air per lb. of oil fuel. As there explained, however, it is impossible to complete combustion with the minimum



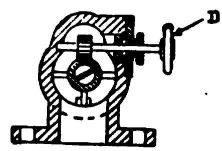


FIG. 26a.

FIG. 26.—KERMODE'S BURNER.

quantity of air, and an excess must be provided, the amount of such excess being largely dependent upon the nature of the fuel, the length of time the gases can remain in the furnace, the direction they follow before release, &c.

Excess of air required.—With solid fuel such as Welsh steam coal burning under good conditions in marine boilers we know that at least 50 per cent. more than the minimum quantity of air required for combustion must pass through the furnaces during the process. With liquid fuel, the excess of air required varies according to the efficiency of the arrangements for spraying the oil and for mixing it with the supply of air. In good examples the excess of air is about 40 per cent. more than chemically necessary, or about 20 lbs. of air per lb. of oil fuel.

Furnace arrangements.—The first requirement, therefore, is the separation of the mass into very finely divided particles, and their dispersion over the whole of the furnace space, which is done by the burner; simultaneously with this pulverising, air must be introduced in sufficient quantity, and so as to ensure its contact with each particle of the fuel before it escapes from the position in which there exists a sufficiently high temperature to admit of chemical combination.

With this object in view the furnace is arranged with means for giving suitable direction to the entering air relative to the direction of the oil streams, and is sometimes fitted with brick bridges or baffles for compelling frequent changes of direction in the products of combustion before passing too far through the space in which combustion should be completed; these brick structures also become reservoirs of heat for the maintenance of suitable temperature, for should the long

column of flame formed by burning petroleum meet comparatively cold bodies, such as boiler plates or tubes, before combustion is complete, further combustion is prevented.

Methods of pulverising compared.—The chief disadvantage of steam as a pulverising medium for marine boilers is the large amount of fresh water lost, which entails the use of an increased evaporating plant. Allowing for the expenditure in the evaporator, the amount of steam used for the nozzle direct would be $3\frac{1}{2}$ to 4 times the expenditure of steam for supplying air by means of a compressor. The amount of steam required to be used in the nozzle may be taken on the average as $\frac{1}{2}$ lb. per lb. of petroleum. Steam is also attended with risk of the flame being extinguished by water carried over with the steam.

Compressed air is free from the objection of loss of fresh water, but it entails the addition of an air-compressing pump, which is an objection where space and weight are limited.

Pulverising by means of the pressure of the oil entails no loss of fresh water and does not require an air-compressor.

The relative conveniences of these systems can only be determined by experience, and the adoption of either will be largely determined by the special circumstances attending the cases to which the system is applied.

There is the same difficulty in each case as regards starting the combustion of the fuel, unless an auxiliary supply of steam is available until steam is raised in the boilers. If this auxiliary steam supply is not available, hand pumps or other special arrangements are employed to raise steam.

Advantages of liquid fuels.—As a fuel for war-ship purposes it possesses many advantages when compared with coal, viz.: (1) Superior evaporation, and therefore reduction in weight of fuel to be carried, or increased radius of action for a given weight. Less space is also occupied by a given weight of fuel. (2) Facility of means for shipping it into the bunkers and transporting it thence to the fires. (3) Reduction of stokehold staff owing to the absence of hand labour in trimming and stoking, which is very exhausting, especially in tropical climates. (4) Less stokehold space required. (5) Absence of coal dust and ashes. (6) No frequent opening of fire doors as when burning coal, hence increased safety with the highly forced boilers of torpedo boats and destroyers. (7) Increased regularity and control over combustion, no reduction of power being necessary to allow for cleaning fires or sweeping tubes, these remaining practically free from material deposit for prolonged periods. The automatic and uniform supply of coal to furnaces which appears from the many abortive trials of mechanical stokers at sea to be recognised as impracticable, is made easy by the use of oil fuel. No fuel need be burnt to waste when the engines stop or variations in steam consumption occur, as immediate alteration or complete stoppage of fuel expenditure can take place, and all waste of steam at the safety valves can be avoided.

Disadvantages of liquid fuel.—(1) The chief objection to the use of liquid fuel at present lies in the uncertainty of obtaining replenishments when required either at home or abroad. (2) Its increased cost as compared with that of coal. (3) For warships the introduction of

further complicated heating, pumping and piping arrangements which are also of appreciable weight, and the loss of coal protection if entirely substituted for coal. (4) Additional risk from fire if the containing bunkers or compartments should be pierced and allow the oil to escape into boiler rooms or bilges, in addition to the loss of the fuel, also greater risk of the generation of inflammable gases. (5) Where steam is used as the pulverising agent the constant loss of fresh boiler water necessitates a large addition to the evaporating plant. (6) Special means are necessary in order to raise steam.

With appliances properly adapted to the particular boilers, skilled management, and fuel freed from objectionable impurities and water, there should be no more smoke than with the best coal skilfully burnt, which is specially important in warships.

Combustion of liquid fuel and coal combined.—In view of the present uncertainty of being at all times assured of obtaining supplies of suitable oil fuel for warships, and the impracticability of providing for the stowage of this fuel in entire substitution for coal in all existing ships, attention has been directed to the economical and other advantages which might attend its use if burned in conjunction with coal.

For this purpose the fire bars remain in place and special furnace fronts are fitted with doors which admit of firing with coal, and burners are applied surrounding the fire door each having an independent air supply.

The principles governing combustion previously described generally apply to this arrangement, and when it is suitable for the particular furnace the results are that not only does the liquid fuel so consumed evaporate 30 per cent. more water than an equal weight of coal, but the coal burnt in combination with it has its calorific value enhanced from the fact that under ordinary conditions when burning coal alone a certain amount of useful combustible is lost with the ashes due to pricking, slicing and cleaning of fires: whereas when burning oil over coal this waste is reduced.

The saving in labour connected with this arduous part of the fireman's work is also considerable, and the advantage is possessed of being able to attain full power when fires are dirty or stokers exhausted, which would otherwise be impossible.

Liquid fuel experiments in the British Navy.—Trials have been made on boilers of various types in many of H.M. ships and have shown the practicability and value of oil fuel burning for warships, and it has been established that in all types of boilers with suitable arrangements there is no difficulty in obtaining with oil fuel the same power as was originally obtained with coal. Success was not at first obtained owing apparently to the limited volume of the space for combustion of the oil in naval boilers as compared with the much greater space available in other boilers proportionately to the amount of oil required to be burned.

CHAPTER VII.

ARRANGEMENT AND EFFICIENCY OF BOILERS—WATER TANK BOILERS.

In this chapter and the succeeding, we will consider the different types of marine boilers in general use, and their efficiencies.

It is obvious that the evaporative power of a boiler must depend largely on the efficiency of its heating surface. The duty of the heating surface is to transmit the heat from the products of combustion to the water in the boiler, and the more of this available heat is transmitted, the greater will be the production of steam. The efficiency of the boiler clearly also depends on the completeness of the combustion as well as on the efficiency with which the heat is transmitted.

The efficiency of the heating surface is the ratio between the quantity of heat transmitted to the water in the boiler, to that available for transmission. The *efficiency of combustion* is the ratio between the quantity of heat available for transmission, and the amount that would be yielded by the complete combustion of the fuel.

The *efficiency of the boiler* is the ratio between the heat transmitted to the water and the total quantity of heat that would be yielded by the complete combustion of the fuel. $\therefore$ The efficiency of the boiler $=$ efficiency of combustion $\times$ efficiency of the heating surface. The efficiency of combustion is dealt with in Chapter IV. The efficiency of the heating surface will be dealt with now.

Efficiency of heating surface.—The conditions on which the efficiency of the heating surface depend are :—

1. Its extent, nature, and condition as to cleanliness.
2. Its position and arrangement.
3. The difference of temperature between the fluids in contact with the two faces.
4. The time allowed for the transmission of heat.
5. The nature of the medium for transmitting heat and the manner in which the heat is transmitted, whether from flame, incandescent fuel, or heated gas.

In comparing the evaporative powers of boilers it is not sufficient to estimate simply the total heating surface, consisting, as it does generally, of furnaces, combustion chambers, tubes, &c., for the powers of transmission of these surfaces differ greatly.

In an experiment made by placing a hot substance in the interior of a cubical metallic box submerged in water, it was found that the upper face generated steam more than twice as fast as the vertical sides, per unit of area, whilst the lower face yielded none. The poor efficiency of the sides was due to the difficulty with which steam

separates from vertical surfaces to give place to fresh particles of water, so that a thin film of non-conducting steam is formed in contact with the plates. By slightly inclining the box, the rate of evaporation of the elevated side was increased, whilst from the depressed side the steam escaped so slowly as to lead to an overheating of the metal.

In an ordinary cylindrical boiler the furnaces above the fire-bars form the most efficient heating surface, next come the tops of the combustion chambers, then the sides and ends, and lastly the tubes, omitting the heating effect of the smoke-box tube plate which is very small.

Use of tubes for heating surface.—By the use in boilers of small tubes, through which the heated gases have to pass, a large amount of heating surface can be obtained in a small space, and this arrangement is necessary, though heating surface in this form is comparatively inefficient.

If we consider the case of an ordinary multitubular boiler, with horizontal tubes through which the gases pass, only the upper halves of the tubes can be considered as effective heating surface, owing to the difficulty with which the steam can detach itself from the lower halves, and also in consequence of the soot, &c., deposited inside the tubes. The direction of the tubes in this type of boiler is the same as that of the currents of hot gases on their way to the funnel, instead of being normal to it, as it should be in order to extract the maximum amount of heat from the gases. Flame cannot pass through long tubes of small diameter, and consequently the useful combustion of the gases cannot extend much beyond the combustion chamber. If the combustion has not then been completed the flame is extinguished within a few inches from the entrances of the tubes, and the gases pass through unconsumed, possibly to burst into flame in the uptake or funnel.

In horizontal tubes the first few inches of length are the most efficient. In coming in contact with the first unit of length, the gases part with some of their heat and proceed at a continually diminishing temperature, as they pass along the tubes, so that only a comparatively small evaporative power can be expected from the exit ends of long tubes, and this is confirmed by experiment.

Experiments on steam-producing power of heating surface.—In 1830, Stephenson found that in a locomotive boiler open to the atmosphere, with the fire-box separated from the barrel, one square foot of fire-box was equal to three square feet of tube surface. In 1840, experiments were made by dividing the barrel of a locomotive boiler into six compartments, that next the fire-box being six inches and the others twelve inches long. These experiments showed that the first six inches of tube surface were equal, area for area, to the fire-box surface, the second compartment was only one-third as effective, whilst in the remaining compartments the rate of evaporation was small.

In 1864 further trials were made on a multitubular boiler five feet long, the tubes being divided into six parts by plates. The compartment next the fire-box was only one inch long, the second ten inches, and the four remaining were each twelve inches in length. The following quantities of water were found to have been evaporated after three hours' work :—

Compartment No.	1 (1 inch long)	. . .	46 ozs.
„	2 (10 „)	. . .	47 „
„	3 (12 „)	. . .	30 „
„	4 („ „)	. . .	22 „
„	5 („ „)	. . .	18 „
„	6 („ „)	. . .	17 „

The high rate of evaporation in the first compartment, which was only one inch long, was no doubt due to the action of the tube plate, but a comparison of the second compartment with the others shows how the evaporative value of the tubes diminished as the gases passed from the combustion chamber to the smoke-box, and gradually gave up their heat and became of less temperature.

Although the exit ends of the tubes are less powerful in steam-producing power than the inlet ends, they still add appreciably to the power of a boiler within the limits of length adopted in practice. In the Devonport experiments on the distribution of temperature inside a marine boiler, the gradual fall of temperature of gases between the two ends of a tube is shown, also the fact that even at the exit end the temperature is still considerably higher than that of the water (Fig. 15).

The steam-producing power of any section of the tube depends really on the difference of temperatures between its inside and outside surfaces, which temperatures are generally unknown. The temperature of the water on one side is however known, as was also, in the experiments referred to, the temperature of the hot gas inside the tube. The steam-producing power will be proportional in some way to the difference between the temperatures of the gas on one side and the water on the other, and this difference even at the end of the tube is, it will be seen by reference to the figure, still considerable.

Diameter of tubes.—For coal that burns with a long flame, the diameter of the tubes should be large, so as to allow the flame to pass as far along the tubes as possible. But for coke and anthracite coal, if the hydrocarbons and carbonic oxide can be burnt before they reach the tubes, they may be made of small diameter, so as to increase the surface and facilitate the action of the hot gases on it.

As a general rule the ratio of length to diameter of tube in marine boilers rarely exceeds 35 to 1, and the area through the tubes should be about one-seventh the grate area. In locomotive boilers the ratio of length to diameter of tubes is often as great as 50 to 1.

For a given description of boiler the evaporative efficiency will depend mainly on the ratio between the quantity of coal burned, and the extent of heating surface to transmit the heat of combustion to the water.

Rankine's formula for efficiency.—Rankine has given the following approximate formula for calculating the efficiency of a boiler :—

Let E be the theoretical evaporative power of the coal.

E' „ its available evaporative power.

S „ the number of square feet of heating surface per square foot of fire-grate.

F „ the number of pounds of coal burnt per square foot of fire-grate per hour.

F

Then the efficiency of the boiler is

$$\frac{E^1}{E}=\frac{BS}{S+AF}$$

Where B and A are constants to be determined by experiment.

The fraction on the right-hand side of the equation, if B be omitted, represents the efficiency of the heating surface itself. B is a fractional multiplier to allow for miscellaneous losses of heat whose value is found by experiment. A is a constant to be found empirically, and is probably proportional approximately to the square of the quantity of air supplied per pound of coal.

Rankine gives the following values for the constants A and B, deduced from the practical performances of a number of boilers.

	B	A
I. The convection taking place in the best manner, either by introducing the feed-water at the coldest part of the boiler, and making it travel gradually to the hottest; or by heating the feed-water in a set of tubes placed in the uptake. Draught produced by the chimney only	1·0	0·5
II. Ordinary convection and chimney draught only . . .	$\frac{11}{12}$	0·5
III. Best convection and forced draught	1·0	0·3
IV. Ordinary convection and forced draught	$\frac{19}{20}$	0·3

When there is a superheater or feed-water heater its surface must be included in computing S.

This formula is framed on the assumption that the losses from imperfect combustion and excess of air are inappreciable, and that the construction and management of the furnaces are the best possible. If this be not the case, the coefficients A and B must be modified to suit the altered circumstances.

Case II., viz. that of ordinary convection and chimney draught, is that of the majority of marine boilers. Rankine gives the value of B in this case as $\frac{11}{12}$, and this appears to agree very well with the actual results of the performances of the high rectangular boilers working with steam pressure of about 30 pounds per square inch. Mr. Robert Wilson gives $\frac{7}{8}$ as the value of B, which seems to approximate more closely to the performances of cylindrical marine boilers. The formula is of little value however.

Low-pressure boilers.—The first boilers fitted to work marine engines were of the rectangular, or 'box' form; an example of this type is shown in Figs. 27 and 28, which represent the most general type of marine boiler for steam pressures not exceeding from 30 to 40 lbs. per square inch above the atmospheric pressure. This form is now entirely obsolete, not being suitable for high pressures, as in these boilers the stresses are resisted principally by the action of straight iron stay-rods. For pressures above 30 or 40 lbs. per square inch, these would have to be so numerous and closely spaced, that the boilers would be excessively heavy, and the internal parts inaccessible.

The furnaces were made with flat sides, and arrangements could be made to keep their crowns sufficiently high above the bars to allow

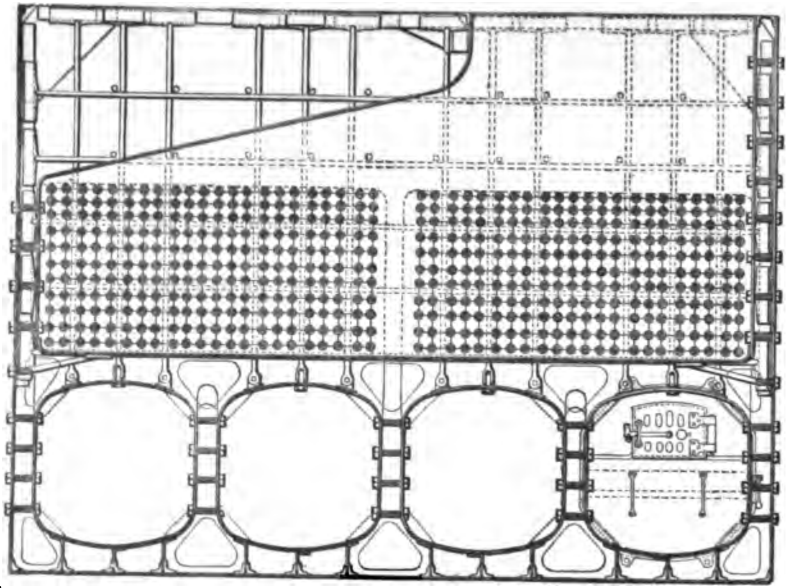


FIG. 27.

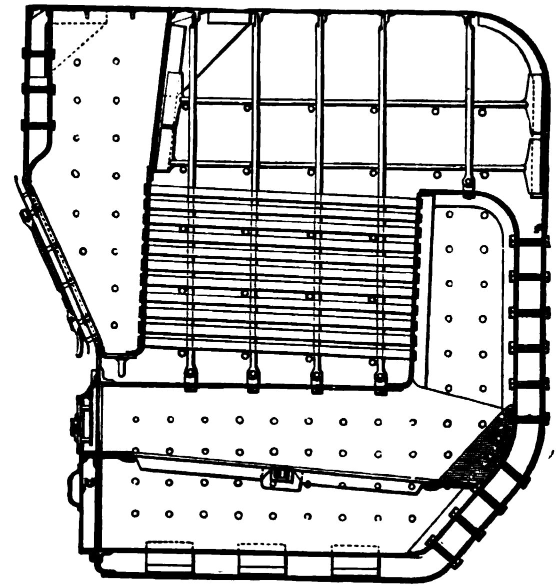


FIG. 28.

the gases to mingle freely with the air, whilst the bottoms of the ash-pits could be kept low enough to permit an ample supply of air to pass through the fires for combustion. Good results were obtained from these boilers, and this was due to a great extent to the very roomy furnaces and combustion chambers with which they were fitted. The furnaces were usually arranged in pairs, each pair having a common combustion chamber, as shown in the diagrams. The object of this is pointed out in Chapter IV., under the heading combustion chamber.

From the combustion chamber the smoke and gases pass through tubes, arranged over the furnaces, to an uptake in the front of the boiler, in which they all unite, and are conveyed to the funnel.

Many boilers of this type were fitted with superheaters, which are described in Chapter XIII. The heat added to the steam in the superheater from the hot gases on their way to the funnel would otherwise have been wasted, and there can be little doubt that much of the saving that has resulted from superheating steam has been due to the partial utilisation of the waste heat in the escaping gases.

The plates in these boilers were thin, the necessary strength being obtained by suitable staying. The area of the stays was made sufficient to resist the action of the whole of the steam pressure, and care was taken to place them sufficiently close together to prevent any appreciable buckling of the plates between the stays when under pressure.

If we take an average of the performances of the best examples of boilers of this type we find that, at full power, about 30 lbs. of coal were burnt per hour, and 10 indicated horse-power developed, per square foot of fire-grate. With boilers fitted with superheaters and the engines with surface condensers, as was usually the case, the consumption of feed-water per hour may be taken at about 26 lbs. per indicated horse-power.

Therefore the quantity of water evaporated per pound of coal was equal to

$$\frac{10 \times 26}{30} = 8{\cdot}7 \text{ lbs.}$$

Taking the theoretical evaporative power of the coal as 14·5 lbs. from 100° Fahr. at 275° Fahr., which is the temperature corresponding to a pressure of 30 lbs. per square inch above the atmosphere, this gives as the actual efficiency of the boiler,

$$\frac{8{\cdot}7}{14{\cdot}5} = 0{\cdot}6$$

These boilers had about 30 square feet of total heating surface per square foot of fire-grate, so that their efficiency calculated by Rankine's formula would be

$$\frac{E'}{E} = \frac{BS}{S + AF}$$

$$= \frac{\frac{11}{12} \times 30}{30 + \frac{30}{2}} = \frac{11}{18} = 0{\cdot}61$$

In this case, therefore, the results of the actual performance agree with those calculated by Rankine's formula.

For higher pressures of steam the rectangular, or 'box' boilers had to be abandoned, and boilers with cylindrical shells and furnaces substituted for them.

High-pressure boilers.—These may be divided into two classes, high or 'return-tube' boilers, and low or 'through-tube' boilers. High boilers are generally used where they can be conveniently arranged for, but the low boilers are fitted in war vessels of small depth of hold to keep them below the steel deck or water line for protection from shot, &c. These boilers are rather inferior to the low-pressure boiler in economy of generation of steam and in the amount of coal capable of being burnt per square foot of grate with the same draught. Figs. 29 and 30 show the general arrangement of a large example of the high type of cylindrical marine boiler used for pressures of 150 lbs. to 180 lbs. per square inch. In the diagram, which represents a four-furnace boiler of about sixteen feet diameter, each pair of furnaces has a separate combustion chamber, which is the most general arrangement in these boilers.

The end plates above the tubes and combustion chamber and below the furnaces are supported by long bar stays passing through the plates with nuts inside, and nuts and stiffening washers outside. The large stiffening washers are riveted to the plate as shown in Fig. 29. The front plate and tube plate between furnaces and tubes are similarly stayed.

The sketch indicates the construction of all the parts in detail. The top of the combustion chamber is stayed by 'dog stays,' and supporting bolts screwed through the combustion chamber plate from below. The bottom of the girder is kept well clear of the top plate to facilitate cleaning and circulation of water. In this boiler every alternate tube in half the horizontal rows is a stay tube (Fig. 35), and these stay the front tube plate to the front of the boiler, while the sides and back of the combustion chamber are stayed to the shell and back of the boiler by short stays screwed through both plates and nutted. The zinc slabs for preventing corrosion are also indicated at various parts of the boiler.

The comparatively low evaporative power and economy in these boilers is mainly due to the form of the furnace. In the low-pressure boilers, with furnaces of approximately rectangular forms, the necessary distance above and below the fire-bars could generally be obtained whatever the width of the furnace may be. But in the high-pressure boilers, in which the furnaces are cylindrical, the height above and depth below the bars, are entirely dependent on the diameter of the furnace, and it is difficult, in most cases, to keep the crown of the furnace sufficiently high above the fires, or to make the ashpit large enough to allow an ample supply of air to the fires. The smaller their diameter, and the greater their length, the greater is the difficulty of securing a proper measure of efficiency.

In these boilers the front itself forms the outer tube plate, and the smoke-box and uptake form an entirely external fitting instead of being partly built in the boiler, as in the case of the earlier low-pressure boilers.

Another feature which tends to reduce the efficiency of cylindrical boilers is the often restricted area of water surface and volume of steam chest, which renders them more liable to priming. *Priming* is the

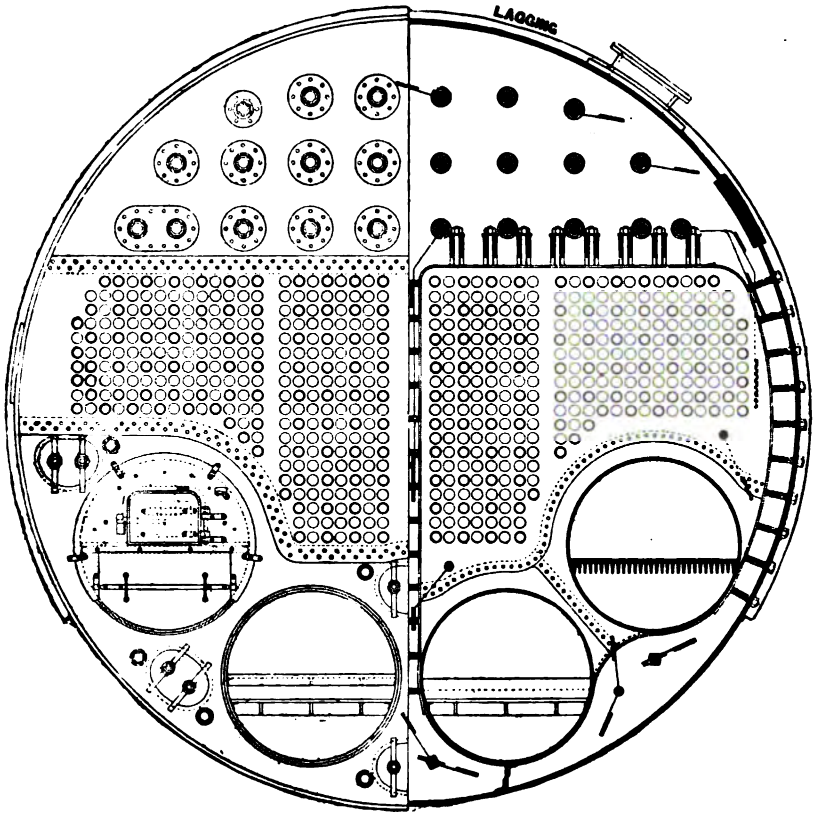


Fig. 29.

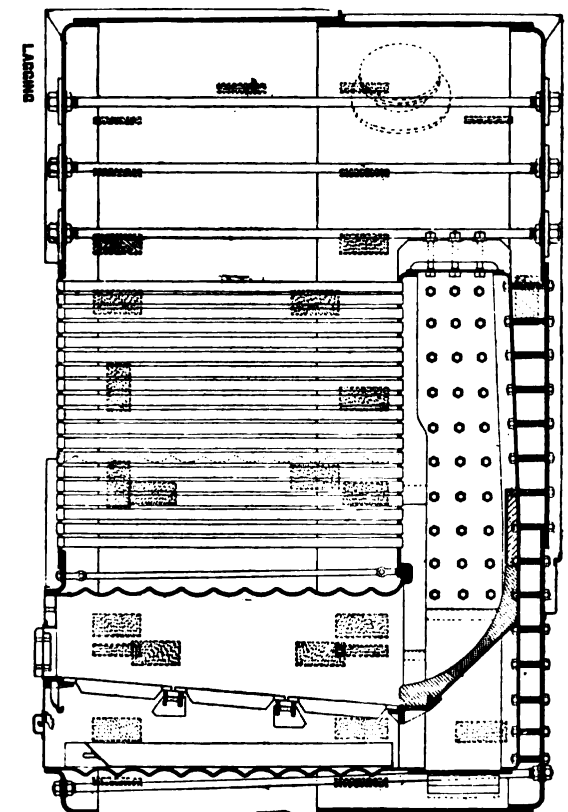


Fig. 30.

name given to the passage of water, with the steam, from the boilers to the engines, which sometimes takes place, and which is liable to produce serious results, reducing the power of the engines, and causing severe stresses on the cylinders, &c., and when it is excessive it may lead to overheating of the boiler plates and tubes, by withdrawing a large quantity of water from the boilers. To avoid this, the depth of the steam chest should, if possible, be at least one-quarter the diameter of the boiler.

Furnace bridges.—The bridge of brickwork at the end of the fire-bars, to limit the extent of the fire, is supported by an iron casting and inclined plate, and the brickwork extends for some little distance up the back of combustion chamber to take the first impact of the flame. The joint of the combustion chamber tube plate with the furnace is generally also protected by a brick lining, as shown in Fig. 30.

Fig. 31 shows another plan of building up this bridge without using any bricks of special shapes, but only the usual square pattern trimmed as required.

The bridges of the older boilers were provided with means for admitting air behind the bridge into the combustion chamber, but this was found not to be efficient for its purpose under usual sea-going conditions, and is not now fitted.

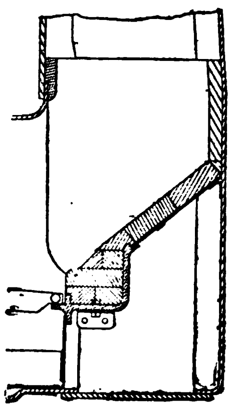


FIG. 31.

Attachment of tubes. Tube ferrules.—The ordinary tubes in boilers are passed moderately tightly into the tube plate holes, and are then rolled or expanded by a roller drift which makes the ends steam and water tight. The stay tubes are screwed into both plates and then similarly expanded by the roller drift to make them tight. Figs. 35 and 36 show the stay tube and plain tube respectively. The diameter of the smoke-box ends of the tubes is made slightly larger than that of the fire-box end to facilitate their entry and removal.

When the tubes are much worn at the fire-box end, they are often supported by inserting and *rolling* into them short pieces of tube termed ferrules. In the Royal Navy all the fire-box ends of the tubes are fitted with the Admiralty 'cap ferrule,' shown in black in Figs. 35 and 36. This ferrule is in contact with the tube, only at some distance away from the plate, and the cap of the ferrule shields the tube plate from the impact of the hot gases, and conducts the heat to that part of the tube away from the joint, where it is transmitted to the water without doing any harm. These ferrules have been very successful in the prevention of leaky tube ends, and in lengthening the lives of the tubes. The cap ferrule is simply driven in tightly.

Low type of high-pressure boiler.—The low type of cylindrical boiler, shown in Figs. 32 to 34, is fitted on board ships of the smaller classes, such as sloops, gun-vessels, &c., and also where, in larger vessels, there is not sufficient room below the vessel's protective deck to enable the high type of boiler to be fitted. Boilers of this type in the Navy have generally given higher evaporative powers than those just described.

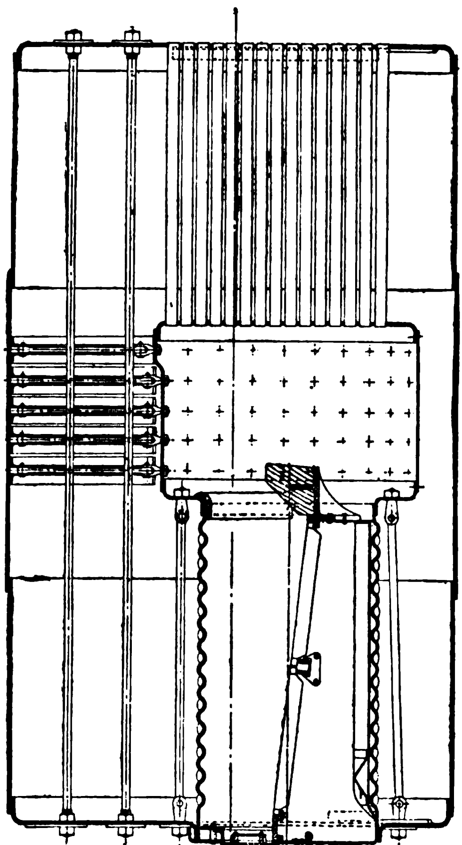


FIG. 33.

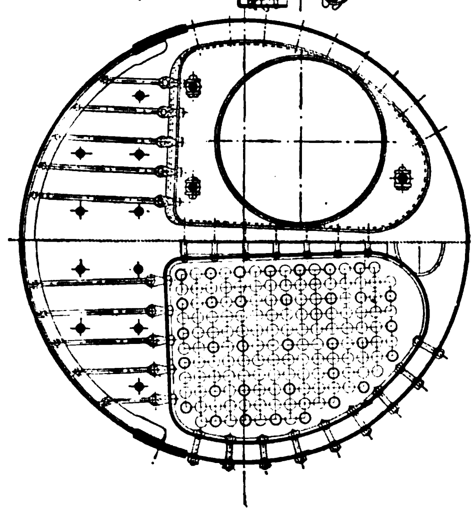


FIG. 32.

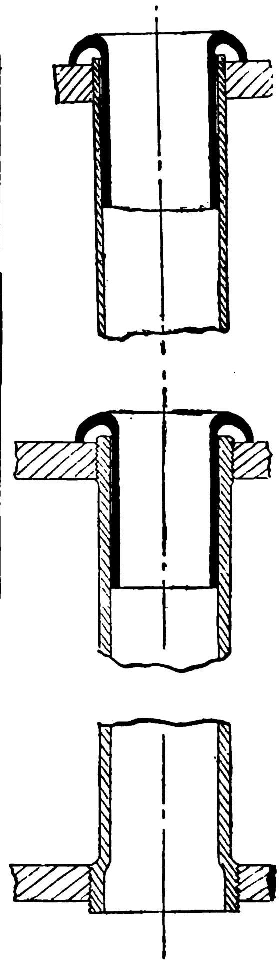


FIG. 36.—Ordinary Tube.

FIG. 35.—Stay Tube.

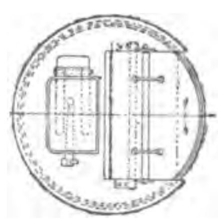


FIG. 34.

The sketches show a low boiler with two furnaces, but in many vessels three furnace boilers of this type have been fitted, and in a few cases four and even five furnaces.

In the large majority of such boilers the furnaces discharge into a common combustion chamber, and such boilers have generally given very good results. The more recent examples supplied for the Navy have been fitted with a divided combustion chamber as recommended by the Admiralty Committee of 1892, so that each furnace has a separate combustion chamber. Fig. 32 shows such a boiler.

In this type it will be seen that the combustion chamber is wide, and that the tubes, instead of returning above the furnaces to the uptake at the front of the boiler as in the high type, are continued along the boiler at about the same level as the furnaces, to the uptake which is situated at the other end of the boiler. The path of the gases to the funnel is thus more direct than with the high type. This type is sometimes known as the 'through-tube type.'

The early examples of this boiler were provided with two fittings which are not supplied to the later examples for the Royal Navy. One of these was a hanging brick bridge in the middle of the combustion chamber, extending from the top to about two-thirds the depth of the combustion chamber, to insure the lower rows of tubes conducting more of the hot gases to the funnel than they probably otherwise would, and thus promote efficiency. Trials showed, however, there was little difference between the efficiency with and without this fitting, and as it is troublesome in practice, on account of frequent renewal, it is not now supplied. The other fitting was the ash-tube, which was, in cases where there was no central furnace, low down in the boiler, fitted below the tubes between the two tube plates, for the purpose of more readily clearing the combustion chamber of ashes, &c. These tubes gave trouble by corrosion and consequent weakness, and were really not required.

Double-ended boilers.—Figs. 37 and 38 show a double-ended boiler. This is a common type for mercantile steamers, and is also fitted in a considerable number of ships of the Royal Navy. It is practically equivalent to two single-ended boilers placed back to back, but lighter for equal power, because the weight of the end plates and of much water in the spaces at the backs of the combustion chambers, is saved.

The arrangements fitted to such boilers as regards combustion chambers have been very varied. In some cases in the Royal Navy, with four furnaces at each end, the whole eight furnaces have been led into one large common combustion chamber. Others have had the combustion chamber divided into four parts by central longitudinal and transverse water spaces, so that each end of the boiler supplies two separate combustion chambers, there being four altogether. This type is illustrated, and is equivalent to two boilers similar to those of Figs. 29 and 30, placed back to back and united. Other examples have had the combustion chambers divided into two parts by a transverse vertical water division, the four furnaces at each end having a common combustion chamber. The most general arrangement, however, is to have three furnaces at each end, and three combustion chambers formed by longitudinal water spaces, each opposite pair of

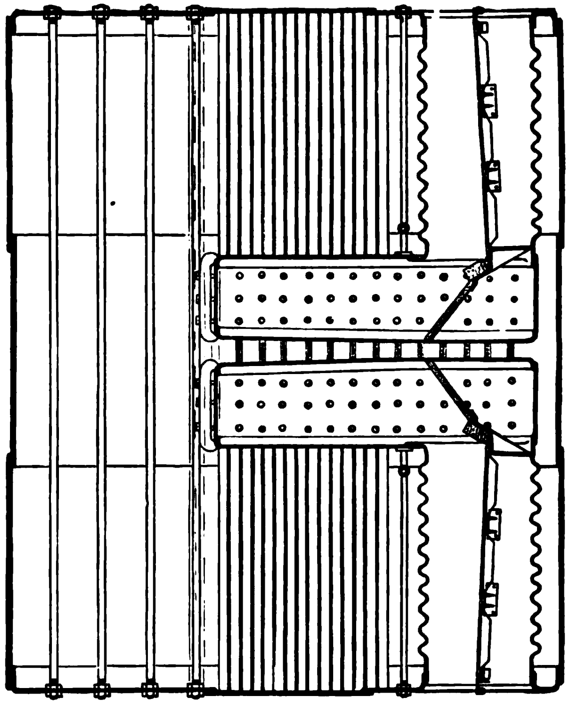


Fig. 38.

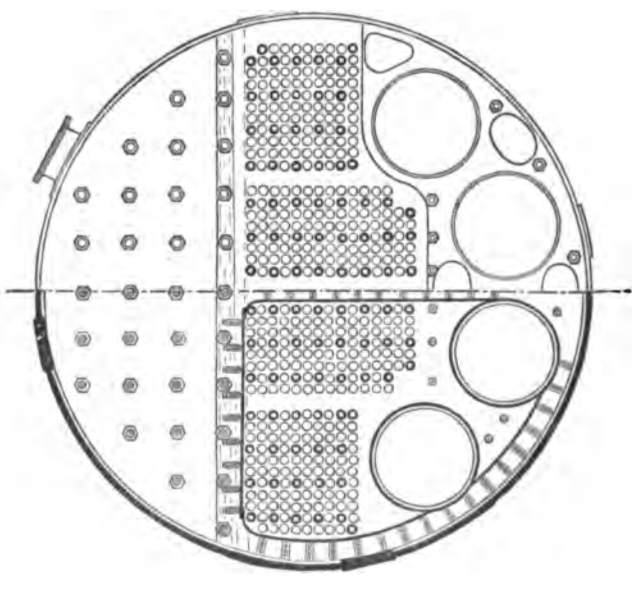


Fig. 37.

furnaces leading to one common combustion chamber. This type of boiler is very common, and has been found, on the whole, to be a convenient and satisfactory arrangement. Several boilers, with three furnaces at each end, have been fitted in the Royal Navy with one large combustion chamber; while again others have had six separate chambers, one to each furnace.

Varieties of furnace tubes.—The early cylindrical boilers had plain furnaces, stiffened at intervals by Adamson joints, such as shown in

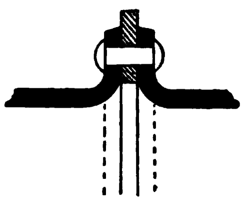


FIG. 39.

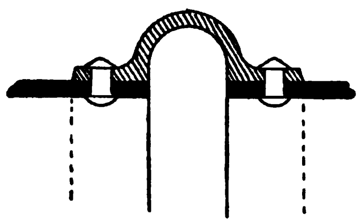


FIG. 40.

Fig. 39, Bowling rings, indicated by Fig. 40, and other devices. These were sufficient for moderately low pressures, such as 60 lbs., without requiring an excessive thickness of furnace plates. Beyond this pressure, however, it became desirable to adopt a stronger form to avoid excessive thickness. Fox's corrugated furnace (Fig. 41) was then for many years almost universally employed for marine furnaces. They were much stronger to resist compressive stress, and enabled the higher pressures to be carried without increase of thickness of plate. Fig. 42 shows the Purves flue manufactured by John Brown & Co., while Fig. 43 shows a later variety of furnace known as Morrison's 'Suspension' furnace.

FIG. 41.

FIG. 42.

FIG. 43.

Marine locomotive boilers.—Figs. 44 and 45 illustrate the locomotive type of boiler which has been used for marine purposes in torpedo boats, &c., in which the working pressures of steam have been from 120 lbs. to 180 lbs. per square inch. In this type of boiler there is a broad and practically rectangular fire-box at one end, the crown of which is strengthened by means of stays to the roof of the boiler, as shown. The example illustrated is by Yarrow & Co.

The air for the combustion of the coal is supplied from underneath, and there is considerable space and height above the fires to allow for the combustion of the gases. The barrel of the boiler beyond the furnace is cylindrical, and contains the tubes which lead to a smoke-box at the opposite end of the boiler. In the cases in which these

boilers have been employed the stokeholds have been closed, and kept under a pressure of air, equal sometimes to 4 inches or 5 inches of water, by means of blowing fans, the rate of combustion of coal per

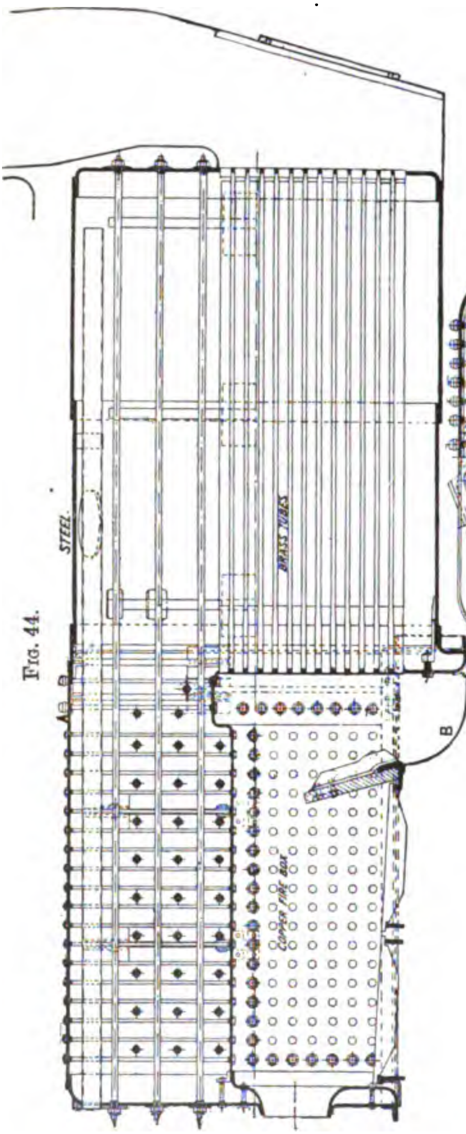


Fig. 44.

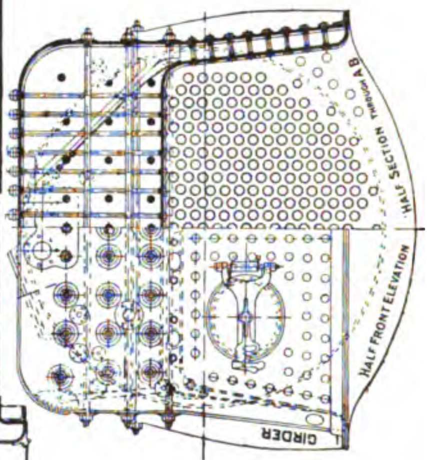


Fig. 45.

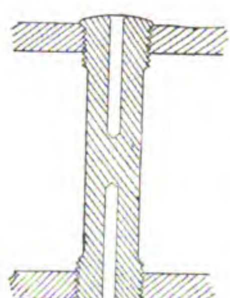


Fig. 46.

square foot of grate being from 80 to 90 lbs. per hour, occasionally reaching 100 lbs.

The fire-box and tube plate are joined to the rectangular shell

around them at the lower part, and the bottom of the furnace is open or 'dry bottomed,' and contains the fire-bars. The tubes are 2 in. diameter in the body, reduced to 1¾ in. for about 9 inches of length at the furnace end and closely spaced, so that the water spaces of the boiler are small, and the weight of the boiler for a given amount of heating surface is correspondingly reduced. All parts of such boilers, except the barrel, require to be closely stayed. The water spaces round the furnace contain large numbers of short screwed stays, either riveted or nutted at the ends Copper stays are often used and these are riveted into place. As such stays are liable to crack inside the water space, small holes are drilled in the stay (Fig. 46) to detect this defect by the leakage through the hole. The furnace roof is stayed by long

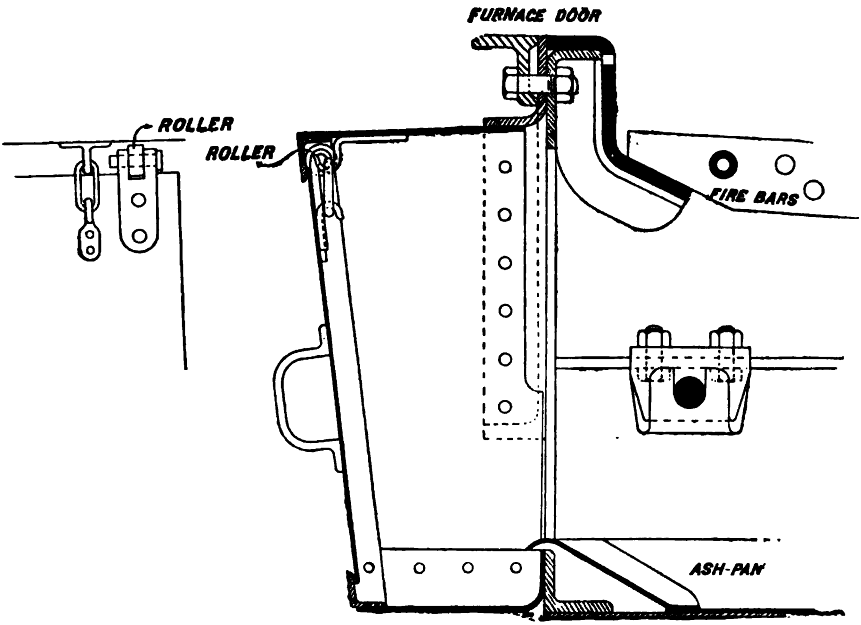


Fig. 47.

stays to the roof of the boiler, and the two ends of the boiler are stayed to each other by long bar stays.

The interior of the boiler, owing to the presence of so many closely-pitched stays, is not accessible for cleaning, so that the use of fresh water with such boilers is essential.

A large amount of heating and grate surface is obtained in these boilers on a limited weight and space, but after a few hours' working at full speed the tubes become choked at the mouths with scoriæ and ash, so that the power then rapidly falls.

A similar boiler, with modifications, has been fitted in many torpedo gunboats. In these the water spaces at the sides of the fire-box have been continued round the bottom below the ashpit, and the

boiler is then called a 'wet-bottom' boiler. The air for combustion in this case enters through the front of the ashpit. In many recent cases the furnace has been divided into two parts by a complete longitudinal water space extending to the tube plate, which is then fitted as two separate plates. The object of these extra water spaces has been to improve the circulation of the water round the fire-box where the heat is most intense, and much weight has thus been added to the boilers, but it is doubtful, after experience, whether any advantage commensurate with the additional weight and cost has been gained.

Automatic safety air inlets.—Locomotive boilers are invariably worked under forced draught, and safety arrangements are provided to prevent injury to the persons in the stokehold in the event of any accident to the fire-box, or any considerable leakage of tubes. Two such arrangements are fitted; (*a*) consists of forming a 'protection box' in the front of the ashpit, through which all the air for the fires

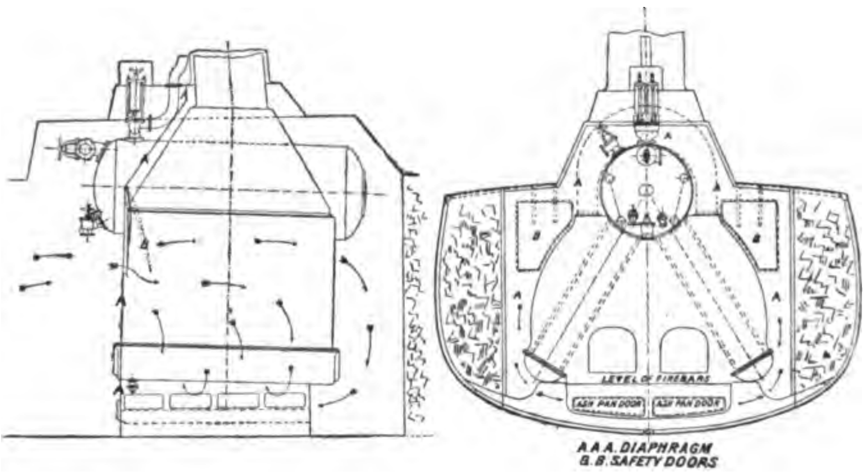


FIG. 48.

has to pass, as shown in Fig. 47. The inlet doors for the protection box are light, hinged or hung at the top, and open inwards. The pressure of air in the stokehold keeps these doors open for the admission of air, but in case of any steam being discharged into the furnace and attempting to enter the stokehold, the protection box doors act as non-return valves, close, and confine the steam and flame to the ashpit, whence it escapes up the funnel.

(*b*) In the second plan the ashpit door is bolted up when under way, a screen or light bulkhead is fitted round the boiler at each end, and non-return air flaps are fitted in the screen adjacent to the stokehold, opening from the stokehold into the space between the screens, as shown in Fig. 48. This space is in free communication with the ashpit, so that while air enters through the flaps when the pressure of air opens them, and proceeds thence to the ashpit and fires, the flaps

would close and prevent any steam or flame issuing into the stokehold if discharged from the furnace. The front of the ashpit is only unbolted for the removal of ashes. This plan (*b*) is evidently only applicable with dry-bottom boilers.

In both torpedo boats and torpedo gunboats, however, the locomotive boiler has now been abandoned in favour of the water-tube boiler described in the next chapter.

Furnace frames and doors.—The furnace frame is generally made double, and an air space arranged between the plates.

In Fig. 48*a*, which shows the door of a Belleville boiler, the door proper has an outer and inner plate, the former being a screen plate, with edges open for the admission of air. The door is perforated with holes at the lower part, through which the air is drawn, and the inner plate, which is of cast-iron, is closed at the bottom, and has holes for the discharge of air at the top. When the fires are alight, there is a continuous current of air flowing into the furnace through these plates.

Fig. 48*b* shows another variety, the air being admitted through holes at the bottom of the wrought-steel door proper, a perforated inner cast-iron plate being fitted to shield the door. The wrought-steel furnace frame which carries the door also has an inner shield plate of cast-iron perforated with holes.

Fig. 48*c* shows the furnace door of a torpedo-boat destroyer. In such vessels, which are heavily forced, all openings connecting the furnaces and the stokehold must be protected by non-return flaps, so that in case of a tube giving way no flame will enter the stokehold. In this case, therefore, the furnace doors are not perforated, but a supply of air is obtained by connecting the tube between fire-door and fire, with the ashpits, a series of holes A (about sixteen in number one-inch diameter) being made in this tube at the upper part, through which the air is forced by the pressure in the ashpit. The door has a series of three or four screen plates lightened with holes, and the air is discharged from the holes into the space between these plates, and is thus warmed and discharged to the fires. This door is a 'self-closing' one, so that, in the event of any casualty happening in the furnace, requiring the stoker to leave the stokehold during the operation of firing, the furnace doors close themselves, and prevent flames entering the stokehold. To effect this, the furnace door is swung on an axis inclined to the vertical, and the lower part of the axis is also shifted out from the front of the boiler. The door, when so fitted, will quickly close by its own weight if released when open.

Influence of workmanship on the durability of boilers.—Unless skill and care be exercised in the manufacture of any structure its intended strength and durability will be decreased. In boiler work great care should be exercised to insure that the holes in the plates at the joints are *fair* before the rivet is put in. If they are found to be not exactly true they should be made so by the use of a rimer, and not be drifted, by which the plate may be actually broken in manufacture, for it is obvious that in such a case durability cannot be expected. Sometimes a smaller rivet has been used in order to get it in and hide the fault. The worst feature of these defects is, that they cannot generally be discovered when the boiler is made, and only show them-

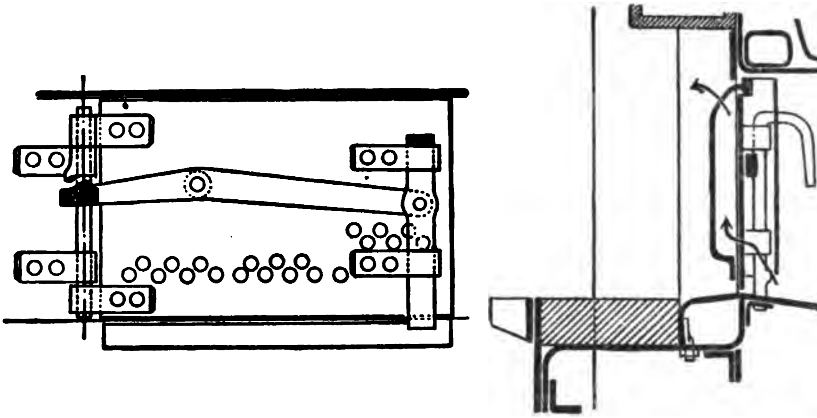


FIG. 48A.

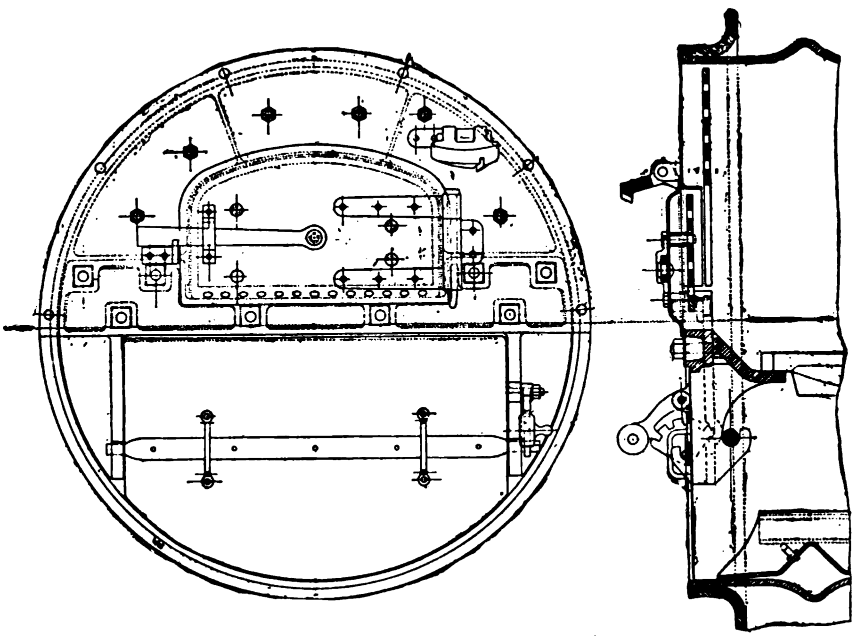


FIG. 48B.

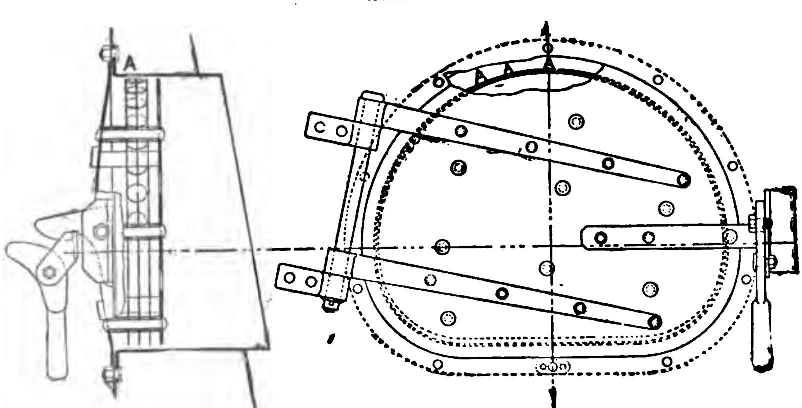


FIG. 48C.

selves after the boiler has been subjected to the stress of actual work, when it is very difficult for them to be effectually rectified.

In good boiler work the rivet holes in the plates are now always drilled. When the plates are drilled together they should be taken apart before being riveted, to allow the burr or sharp edge to be taken off from the holes.

Another point to be secured is to get the joints properly closed, so that little caulking is necessary. In good boiler-making, the joint should be tight without caulking; and if it be not fairly tight no amount of caulking will permanently remedy it, though it may conceal the defect for the time. Excessive caulking is very injurious, and is probably one of the most fruitful causes of the *grooving* that sometimes occurs along the rivet-seams. It also tends to raise the edge of the upper plate and cause looseness at the joint. The edges of the plates should be planed with a bevel of about 75°, and the only caulking required should be a little along the thinner edge of the bevel.

CHAPTER VIII.

WATER-TUBE OR TUBULOUS BOILERS.

THE previously described boilers, having the water outside the tubes and contained in an outer shell or so-called tank, are technically called 'water-tank' boilers, to distinguish them from the class known as 'water-tube' boilers described below.

'Water-tube' boilers are those in which the flame and water in the older form of boiler are interchanged, so that the water being evaporated is contained inside the tubes, and the hot gases outside them. The hot gases outside the tubes are confined and led to the funnel by a casing fitted for the purpose.

The desire to obtain boilers having the capacity of safely generating steam of higher pressures than had been previously used, combined with lightness of construction, and having the tube ends favourably situated for resisting leakage, has led engineers for many years to seek for a satisfactory water-tube boiler.

History of marine water-tube boilers.—The earlier examples fitted in the mercantile marine, commencing on the Clyde about 1857, were not successful; they generally failed owing to rapid corrosion of the tubes, combined in some cases with incrustation due to saline deposits on the water side of the tubes, from salt water which leaked through condensers or was admitted to supply the waste of feed-water. This incrustation was usually not readily accessible for removal.

Most of these early water-tube boilers were eventually removed and replaced by the cylindrical multitubular boilers previously described, the pressure of steam being correspondingly reduced.

Later on, about 1870–75, with higher pressures, renewed attempts were made in Great Britain to obtain such boilers, but they were again unsuccessful, and for some years after this, the attempt in this country was practically abandoned. Mr. Loftus Perkins was perhaps the most successful, and the Perkins boiler and engine, with pressures of 300 to 500 lbs. per square inch, attracted considerable attention. Other of these early boilers were the Rowan, Howard, and Root types.

In France an important application of such a boiler was made in 1879, by the fitting of Belleville boilers to a despatch vessel which was employed on service to a considerable extent, and her boilers gave satisfaction, so that from this time there was a gradual extension of the use of this type in the French Navy. A cruiser launched in 1885 was the next vessel fitted with these boilers, followed in 1889 by another, of 8,000 I.H.P., and two torpedo gunboats. Subsequently practically all new French war ships have been fitted with water-tube boilers, a large proportion being of the Belleville type.

The Messageries Maritimes Company have also largely used Belleville boilers, and as the result of experience this company has fitted these boilers to all their new passenger vessels.

A considerable number of vessels of the French Navy have also been fitted with water-tube boilers of the Niclausse, and also of the Lagrafel-D'Allest type, but the latter type has been abandoned for new French warships. During recent years large warships of the French Navy have been fitted with either the Belleville or Niclausse type.

In England the Thornycroft water-tube boiler was fitted in 1882 to a mission steamer, and in 1885 to a second-class torpedo boat. They have since been applied to a large number of British and foreign war vessels, principally those of small size and high speed. The Yarrow water-tube boiler has also been largely used in warships of all classes. At the present time all the smaller warships and many large ones of the British Navy are being supplied with Yarrow boilers. In 1893, Belleville boilers were ordered to replace defective locomotive boilers of the 'Sharpshooter.' Subsequently the 'Powerful' and 'Terrible,' large cruisers each of 25,000 I.H.P., were supplied with them, and since then a large number of new vessels for the British Navy have been supplied with similar boilers. They were discontinued after the report of an Admiralty Boiler Committee which recommends the tests of other types of boilers.

In England the Babcock and Wilcox water-tube boiler, previously fitted in some small vessels, was fitted in 1893 in the s.s. 'Nero,' and other mercantile vessels have since been fitted with this type, both in Great Britain and the United States; and a large number of battleships, cruisers and sloops have been, and are being, supplied with them both in the British and United States navies.

Niclausse and Dürr boilers have also been fitted in several large ships in both the British and United States navies, while in the German navy large numbers of Dürr boilers have been fitted.

Comparison with locomotive type.—As regards the water-tube boilers which give the greatest power for the weight and the space occupied, a considerable extension took place in Great Britain in 1893 by the demand for small vessels of high speed known as 'torpedo-boat destroyers.' Many varieties have since been tried in such vessels with satisfactory results, and they have entirely superseded the locomotive type, which had previously been used for such purposes. An example of the results obtained in sister vessels with locomotive and water-tube boilers may be given in the trials of 'Havock' and 'Hornet,' built by Yarrow & Co., the 'Havock' with locomotive boilers having copper fire-boxes similar to that illustrated in Fig. 44, and the 'Hornet' with water-tube boilers similar to Fig. 70.

Ship	Type of boiler	I.H.P.	Speed in knots	Weight of boilers, mountings, brickwork, and water, tons	Grate surface, sq. ft.	Heating surface, sq. ft.
'Havock' .	Locomotive	3,497	26·18	43	100	5,010
'Hornet' .	Yarrow's water-tube	3,884	27·6	44·8	172·8	8,216

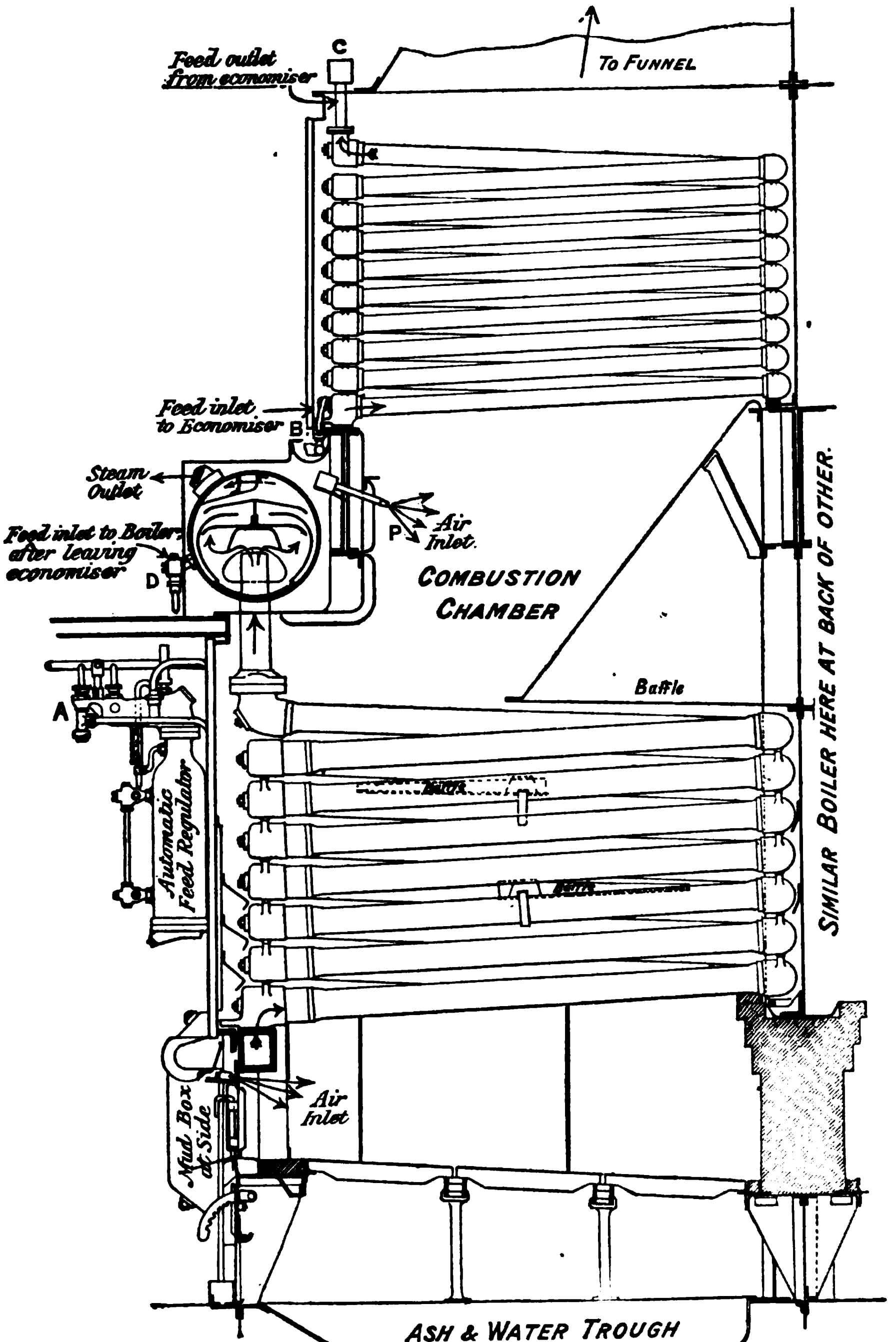


Fig. 49.—Belleville boiler with Economiser.

The advantages of these boilers are:—Their lightness for the power generated, the capacity of raising steam quickly owing to the small quantity of water carried, and their comparative freedom from leaky tubes, the joints being more protected from the direct impact of flame.

The Belleville boiler. Economiser type.—This is shown in Fig. 49. It consists of two distinct series of straight tubes of comparatively large diameter, viz., a lower series forming the boiler proper and termed the 'Generator,' and a smaller upper series forming a feed water heater and termed the 'Economiser.' The feed water is passed first through the economiser, where it is heated by the gases passing to the funnel; it then passes to the lower tubes, where steam is formed.

The Generator consists essentially of a top steam cylinder and a lower water chamber, with a series of straight zigzagged tubes connecting them. There is an external return water-pipe on each side, connecting the ends of the top steam chamber with the lower water chamber. The generator tubes are placed above a rectangular brickwork furnace and enclosed in a sheet steel casing, which confines the flame and gases generated from the coal. A series of *baffle plates* are secured at intervals among the tubes, to insure that the gases should

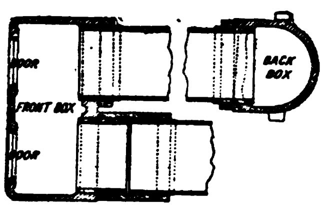


FIG. 50.—Section along Tubes through Upper Boxes.

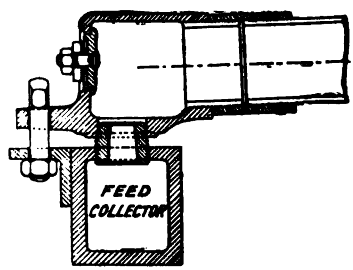


FIG. 51.—Vertical Section through Lowest Box.

traverse the whole of the surface of the tubes, in order to obtain the full value of the whole of the heating surface.

The generator tubes are arranged in vertical groups, termed 'elements,' forming a flattened spiral, the tubes being straight, generally of $4\frac{1}{2}$ inches external diameter, and about 7 ft. 6 in. long, with the ends connected by being screwed into malleable cast-iron boxes, which form the turns of the spiral (see Figs. 50 and 51). The inclination of the tubes is about $2\frac{1}{2}°$ to the horizontal, and there are generally fourteen tubes in each element.

The tubes in each element form one continuous passage for steam and water from the bottom to the top, so that the water, after having passed through any tube, has to traverse a short distance horizontally through an end box before entering the next tube in its ascent.

The front boxes have hand holes opposite each tube, enabling a light to be passed to the end of the tube, to enable the condition of the interior to be seen, and its cleaning effected. The malleable boxes are shaped to allow brushes or scrapers to be inserted to enable the

outsides of the tubes to be cleaned. Both inside and outside of the tubes are therefore accessible for thorough cleaning and inspection from the stokehold floor. A steam or compressed air tube-sweeping apparatus is also provided for use when under steam. This consists of a flat nozzle which can be inserted between the end boxes and enables the soot and scoriæ to be blown off the tubes.

The front of the lowest part of each element is connected to the water chamber, and the front of the top of each element is connected to the steam collector. The connection between the element and the water chamber is made by means of a coned nipple secured by a single bolt (see Fig. 51). The inlet orifice through the nipple is reduced considerably below the area of the tube, to insure that the central elements shall obtain their fair share of the water brought by the return tubes. The amount of reduction necessary is found by experiment, by the insertion of small pipes in the top tubes with cocks outside, and observing the proportion of steam to water discharged from them.

A series of 7 to 10 elements are placed side by side inside the casing. The flame or gases traversing between the tubes generate steam, and cause a circulation of steam and water from the lower water chamber through the tubes, and the mixture is discharged into the top steam collector, where a series of baffle plates are fitted for separating the steam from the water, the steam being drawn off through the stop-valve, and the water flowing along the bottom of the collector to the return water pipes on each side, and thence again to the elements.

The Economiser consists of a number of elements one or two less than contained in the generator, but of smaller diameter, viz., 2¾ inches instead of 4½ inches, placed 4 or 5 feet above the top generator tube in the boiler uptake, and through which the feed-water is first discharged and heated before being admitted to the steam collector. The number of tubes in each element varies from 12 to 20, and their length is generally about 6 ft. The economiser is formed similarly to the generator, and consists of straight zigzagged tubes screwed into malleable cast-iron boxes. The fronts are fitted with cleaning doors, and the casing is provided with doors for access to the tubes as in the generator.

Course of the feed-water.—The feed-water proceeds from the pump to a valve A worked by a feed regulator, thence through a non-return valve to the pipe B, called the 'cold water collector,' at the bottom of the economiser, and then enters the tubes through an orifice in each lower front box, and is pumped upwards to and fro through the elements, abstracting heat from the gases. It issues into a pipe C, called the 'hot water collector,' at the top of the economiser, is thence led to the non-return valve D at the middle of the steam collector above the generator; it then falls to the bottom of the collector, and flows to the return water pipes with the water emerging from the tops of the elements. The tops of the elements are carried up to about eight inches above the bottom of the collector to prevent this water entering the element and interfering with the continuous discharge of steam and water therefrom.

Sediment chamber.—Before entering the lower water chamber

from the return pipe the water passes a non-return valve to a sediment collector at the bottom of each return pipe, shown in Fig. 52. This non-return valve prevents the water leaving the element and ascending the return tube when the vessel is rolling, and regulates the circulation, especially when steam is being raised. The resistance to the motion of the water caused by the tubes is considerable, and on raising steam there is often a tendency for the heated water of the lower tube to flow into the return pipe, causing reversal and confusion of the currents.

The sediment chamber has a division plate, and the lowest part forms a fairly large space in which the velocity of the water is not great, so that nearly all the grease, lime, or other solids settle to the bottom and do not enter the tubes. The bottom of the chamber is provided with a double blow-off valve, the deposits being blown away about once a watch.

The impurities which may be contained in the feed water are the oil from the engines which passes the feed-water filters, and lime sulphate and other salts, due to leakage of salt water at the condenser tubes, or, in emergency, by the admission of sea-water to make up losses, also any undissolved lime that may have passed through the lime tank.

The intentional admission of sea-water for feed water make-up should only be resorted to in cases of emergency, such as the failure of the fresh-water appliances at a time when a continuation of steaming is imperative, but the admission of small quantities of sea-water due to leaky condenser tubes is common, and a small quantity of grease must also be anticipated. The arrangements described keep most of these impurities out of the generating tubes of the boilers, where they would be a source of danger.

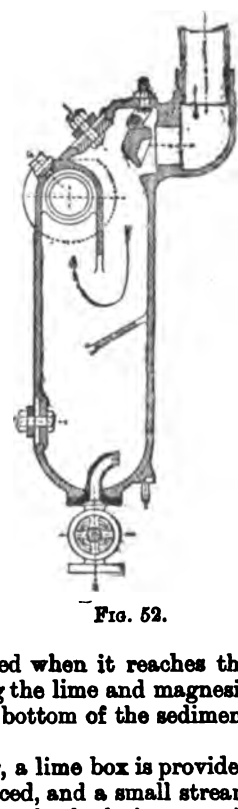


FIG. 52.

The temperature of the feed-water is so raised when it reaches the sediment collector, that it is incapable of holding the lime and magnesia salts in solution. They are precipitated to the bottom of the sediment chamber, whence they can be blown out.

To neutralise any acidity in the boiler water, a lime box is provided in the engine-room in which lime hydrate is placed, and a small stream of water is admitted to it at a high pressure from the feed-pipes to stir up the lime and facilitate its solution or mixture. It is then led to the feed-tank and pumped into the boilers. The lime also facilitates the deposition of grease contained in the feed-water, which is more difficult to deposit than sulphate of lime. Very little lime or grease should be found in the boiler tubes, even with considerable sea-water feed make-up, provided the arrangements are properly used. Practically all grease that enters is found in the lowest row.

Belleville automatic feed gear.—A very important part of the

Belleville boiler is the automatic feed apparatus, by which the water is maintained at its working level.

The apparatus (Figs. 53 and 54) consists of a chamber connected at the top and bottom with the elements of the boiler, and to which one of the gauge glasses is attached. This chamber contains a hollow float. The feed pipe is carried to the chamber, and a valve and spindle A is placed on it, connected with the float by levers and weights so that as the water rises and lifts the float, the weights at the end of the lever close the valve. The admission of feed water then stops until the water in the chamber falls sufficiently to enable the float to descend and raise the weights and open the valve. The feed water is led from this valve to the non-return valve at the economiser. There is no connection between the feed pipes and the regulating chamber, although the feed-valve at A is bolted to this chamber for convenience.

Particular care is taken as regards accuracy of the fitting of the various parts, knife edges being fitted for all bearings to reduce friction, but even so the float, before it can open the valve, has to overcome the friction of the rods at B and A working through steam and watertight glands respectively, and when closing the valve the weights have to overcome this friction. As the float has to be made of substantial strength to withstand the external water pressure, it is not light enough to be water-borne with about half its volume immersed, as it should do to enable proper feed regulation to take place in each direction. To enable it to float freely with about half its length immersed, the weights C are added outside. They have the effect of altering the position at which the float and connections are water-borne. Most of these weights are fixtures, but a number are portable, for use as explained below. A handle is provided to move the gear and ascertain if it is free and working correctly, and the spring shown below the weights is provided to return the lever to its proper position when the handle is released after being pulled down. It also reduces jerks due to the alteration of pressure at the ends of the stroke of the feed pump. Great care should be exercised in packing the glands of the rods of this apparatus, so as to reduce friction. A special form of anti-friction packing is used, composed of about 40 per cent. of tin with a little antimony, 46 per cent. of lead, 10 per cent. of graphite, and 4 per cent. of mineral oil.

To keep the float and gear of moderate dimensions, the automatic feed valve has to be limited in size, and, in consequence, a considerably higher pressure in the feed discharge pipe is required than exists in the boiler. For a boiler pressure of 250 lbs. per square inch, the pressure delivered by the feed-pumps on exit from the latter is about 450 lbs.

Fusible plugs.—A safety arrangement is fitted to each element in the shape of one or two fusible lead plugs driven into small holes bored in front of the boxes. They give warning should there be overheating due to any obstruction to the free entry of water or any failure of the feed-water supply.

Amount of water carried.—Experiment has shown that at ordinary rates of evaporation sufficient water is present in the generator tubes of the boiler when the water in the glass is about midway between the two points of attachment of the gauge pipes to the element. The middle of the water gauge is therefore placed at this point, and the

mechanism so arranged that the water level at ordinary rates of evaporation is kept approximately at $\frac{1}{2}$ to $\frac{5}{8}$ glass. It should be noted that, unlike the water-tank boiler, the height of the column of water in the gauge glass does not simply indicate the amount necessary to

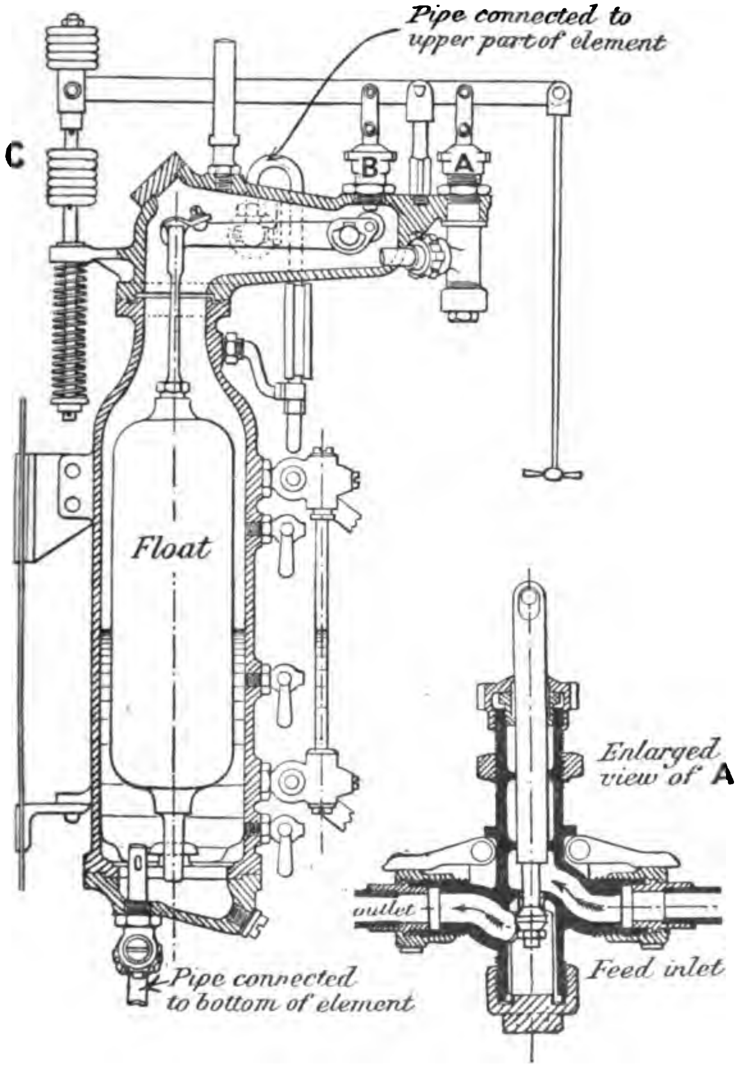


FIG. 53. FIG. 54.

balance the statical pressure of a similar column of water in the boiler, for there is a small difference of pressure between the upper and lower points of attachment of the gauge connections, due to the force necessary to cause the flow of water and steam through the element.

The height of the water in the gauge glass therefore indicates the weight of the water between the points of attachment plus the pressure necessary to cause the flow of water and steam, and as this latter increases with the rate of combustion, the height necessary in the gauge glass to carry the same quantity of water in the boiler increases with the rate of evaporation.

Action when stopping or starting.—At the higher rates of evaporation, more water should be carried in the gauge glass, so that a few portable weights are supplied on the feed regulator, by removing which, the float sinks lower in the water, so that a higher level is required in order to shut the feed-valve. If the evaporation be suddenly stopped the mixture of steam and water subsides, and the level shown by the glass gauge will fall until it corresponds with this. As, however, any lowering of water level causes the feed-valve to open, and thus reduce the pressure in the feed discharge pipe, the feed-pump immediately increases speed, and pumps the water level back again to the shutting-off point, at about ½ to ⅝ glass. This lowering of level on stopping is to be expected, and does not indicate any defect. The feed-pumps will continue to work till the water level is restored.

Conversely, when getting under way quickly, as the feed-pumps will have pumped the water level back to about ½ or ⅝ glass, whereas when at work the quantity required is less than is contained in the boiler under these circumstances, the excess of water is passed into the steam pipe to be caught at the separator, or if the increase of speed be made slowly, it is gradually evaporated, the feed inlet-valve closing while this is happening. This explains the usual phenomena with these boilers—i.e. feed-pumps increasing speed when slowing down, and working slowly when the speed is quickly increased.

Gas-mixing appliances.—The space between the fire and the lowest tubes is not large, so that unless complete combustion of gases takes place within a short distance of the fire their combustion will take place above the lowest tubes, and some heat will pass off, which, had the combustion taken place lower, would have been abstracted by the water in the generator tubes. With a properly regulated fire which passes the proper amount of air there is less likelihood of this, but it has been found that the forcing of air in small streams under pressure above the fire has a beneficial effect in thoroughly mixing the gases, and so accelerating their combustion. The apparatus is simply a gas-mixing one, and not for forcing the rate of combustion of the coal.

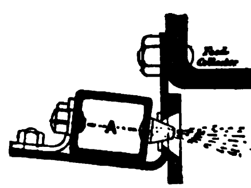


FIG. 55.

The arrangement consists of a square air-pipe A, Fig. 55, about 10 or 12 inches above the fire-bars, into which air is forced in small jets by an air compressor at a pressure of 5 to 15 lbs. per square inch. A series of nozzles about $\frac{3}{16}$-inch diameter, slightly inclined downwards, discharge the air among the gases above the fire with such force as to reach all parts of the furnace and thoroughly mix the gases and oxygen. In the front of the boiler casing are two or three sight holes, with sliding covers, which enable the condition of the fire to be inspected

and the admission of air regulated by experience, so as to obtain the best result.

Combustion chamber.—The space between the generator and the economiser acts as an additional combustion chamber, in which any unconsumed gases that may have passed the lower parts of the boiler may be provided with additional air to assist their combustion, prior to traversing the spaces between the economiser tubes and giving up heat to the feed-water contained therein.

A few small nozzles P (Fig. 49) are therefore fitted to supply compressed air from the furnace air pumps to the combustion chamber to supplement that admitted by the gas-mixing jets above the fire. A small safety valve is fitted on the discharge pipe from the economiser.

Facility for repairs.—Pipes are screwed into the steam collector to the height previously explained, as shown in Figs. 49, 56, and 57, and the tops of the elements are secured to them by flange joints. The bottom of the element is secured to the feed-box or collector by means of a conical joint, shown in detail in Fig. 51, the jointing material consisting of one or preferably two, thin conical nickel rings inserted between the coned nipple and the conical hole in the lower box. The single 'anchor bolt' shown is all that is necessary for satisfactorily making this conical joint.

Should it be found necessary to renew any tube, this can be accomplished by withdrawing the 'element' containing it from the boiler. The 'anchor bolt' which makes the joint with the feed collector at the lower part, and also the four bolts making the flanged joint of the top tube with the steam collector are removed, the front of the element is then lifted sufficiently to allow the lower box to clear the cone of the feed collector, sufficient spring existing at the top tube of the element to enable this to be done. The element is then free and can be drawn out to enable the new tube to be fitted.

Belleville boiler without economiser.—The early Belleville boilers have no economiser, but only the generator portion, this consisting of elements of exactly the same construction and with the same fittings as in the previously-described boiler, except that the height is greater, each element containing twenty tubes instead of fourteen. The feed-water, after leaving the feed regulation valve, proceeds direct to the non-return valve on the steam collector.

Admiralty Boiler Committee.—Owing to the diversity of opinion as to the relative merits of various boilers the British Admiralty appointed a committee in 1901 to investigate and report on the various types. In the preliminary report of this body they recommended that either or all of the following should be fitted in British warships, viz., Babcock and Wilcox, Niclausse, Dürr or Yarrow. In their final report they confined their recommendation to the Babcock and Wilcox or Yarrow large-tube types for large warships, and since their report these two types have been fitted in about an equal number of large warships to the exclusion of other types.

Babcock and Wilcox Boiler.—One of the best known water-tube boilers on land in England and America is the Babcock and Wilcox, and it is also extensively used in both countries for marine work. It is one of the earliest of such boilers, dating from the year 1868, but was almost entirely used for land purposes until 1889, when the

first boiler was constructed for a small vessel. The development of this boiler since for marine purposes has been rapid.

It is simply and strongly constructed entirely of forged steel and without stays, and requires no special training or experience to manage it. The tubes are plain and straight, without threaded, coned, or reduced ends, and can be quickly replaced in a boiler with new ones. A single tube can be replaced without disturbing any others, and it can be readily fitted together on board the vessel, so that large deck openings are not required either for first fitting or subsequent renewal.

Owing to its rectangular form, it fills economically the space available and provides a large grate area and heating surface without placing the tubes too closely together. This large grate area favours the maintenance continuously of a large proportion of the full power of the ship, since if small, the fire grates become dirty much more rapidly, and cleaning has to be more frequently carried out. The larger grates, too, require less forcing, hence less air pressure, which favours cleanliness of the ship, comfort for the men, and efficiency generally.

Details of construction.—In this boiler (Figs. 56 and 57) the generating tubes are fitted between a number of headers, or narrow sinuous vertical water chambers of square section, each pair of which (one at the front and one at the back) is united by tubes inclined at an angle of about 1 in 4. The tubes are expanded, and are examined, through doors in the headers, there being doors in both front and back headers and space being allowed at the back of the boiler for access to the latter. The front and back headers are connected by short and long pipes respectively, as shown, to the cylindrical receiver above them, which is kept, when at work, about half-full of water. The gases from the fire pass around the tubes and thence to the funnel, and the water circulates up the inclined tubes to the upcast headers A, thence to the long top pipe to the receiver, and the downtake headers B, back to the generating tubes. The circulation is good and every section is independently provided for.

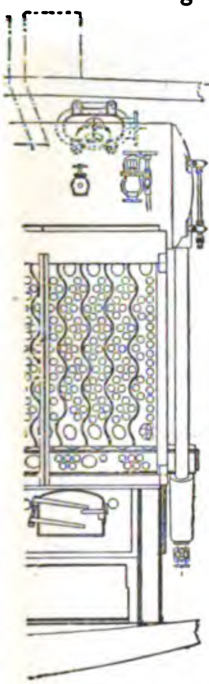
The tubes are expanded into the headers by ordinary roller expanders. The bottoms of the headers are connected by horizontal square pipes, between which, in the type shown in Figs. 56 and 57, run the lowest of the boiler tubes. The square pipe connecting the lower ends of the downtake headers has a sediment chamber at each end, and to this the blow-out apparatus is fitted. There are return tubes at the ends of the cylindrical steam receiver leading to the bottom of the headers. The usual brickwork furnace is constructed below the tubes.

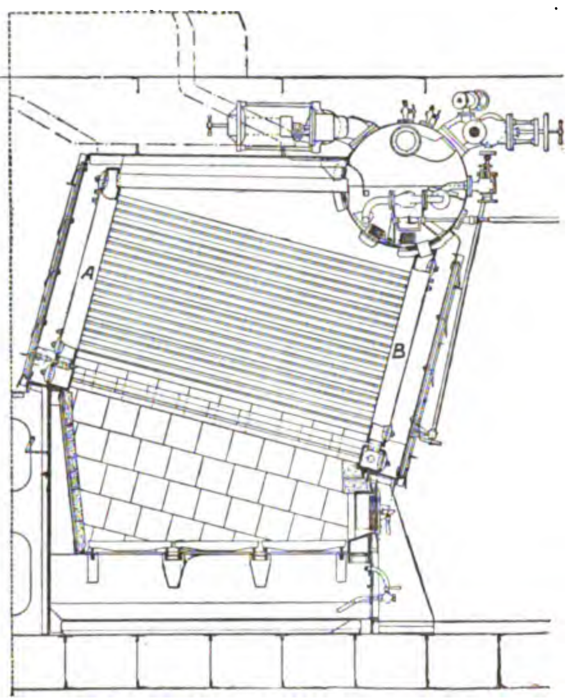
Variation of Details.—The details of boilers of this type vary considerably. In some varieties there are additional vertical tubes, forming the casing of the boiler, and placed on each side. The practice also as regards size of tubes varies. In some cases all the tubes are of fairly large diameter, about $3\frac{1}{4}$ inches, while in others the upper tubes are smaller ($1\frac{5}{8}$), the lower ones only being of large diameter ($3\frac{5}{8}$). In some cases the tubes are of the smaller diameter throughout.

Baffling of gases.—In some boilers of this type the gases pass straight to the funnel as in Fig. 57, without having baffles fitted among the tubes, but the efficiency of Babcock and Wilcox boilers, as of water-tube boilers of most types, is considerably increased by fitting baffle-plates in suitable positions among the tubes, by which means the hot gases on their way to the funnel are compelled to pass over the whole

of the heating surface of the boiler, and thus to give up as much as possible of their heat. In these boilers various arrangements of baffle plates have been fitted. One arrangement is shown in Fig. 58. Another, shown in Fig. 59A, is known as 'wheel baffling,' and is found by experience to be the most efficient and is fitted in the most recent boilers of this type. The result is that the uptakes and funnels do not become overheated even when the boilers are being forced, and the efficiency of the boiler is very high.

Latest type of Babcock and Wilcox boiler.—The most recent type is shown in Figs. 59A and 59B. In this case the two lower rows of tubes are of larger diameter and have each a separate door for access in





Babcock & Wilcox boiler.

FIG. 56. FIG. 57.

the header. The remaining upper tubes are of smaller diameter and are arranged in groups of four, each group of four tubes being accessible through a single door in the header. It will be seen in Fig. 59B that the side casings are well protected by a vertical row of tubes, the side headers being specially formed to allow of this, so that the durability of the casings is thus made satisfactory. The lower tubes are not attached to the square connecting pipe as in Fig. 56, but direct to the headers, as in the case of the upper tubes. The single return pipe between the steam collector and the top of the back headers is replaced by two such pipes to give freedom for escape of the mixed steam and

water circulating from the top of this header. A vertical row of cleaning orifices is supplied in the side casings for each division into which the vertical baffles divide the tube length, one cleaning orifice between each horizontal row of tubes, as indicated in Fig. 59A.

Feed heating and superheating.—A series of tubes are sometimes fitted in the uptake above the generating tubes, through which the feed water passes, forming a feed water heater, while superheaters are also sometimes fitted just above the tubes, as in H.M.S. 'Britannia,' where a portion of the boilers are so fitted. Superheaters have also been supplied in many cases in the mercantile marine.

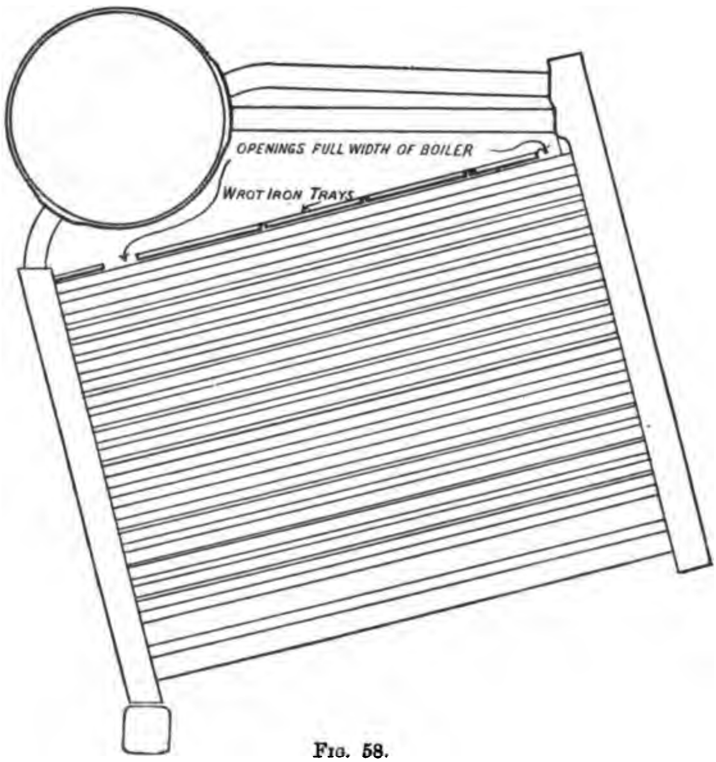
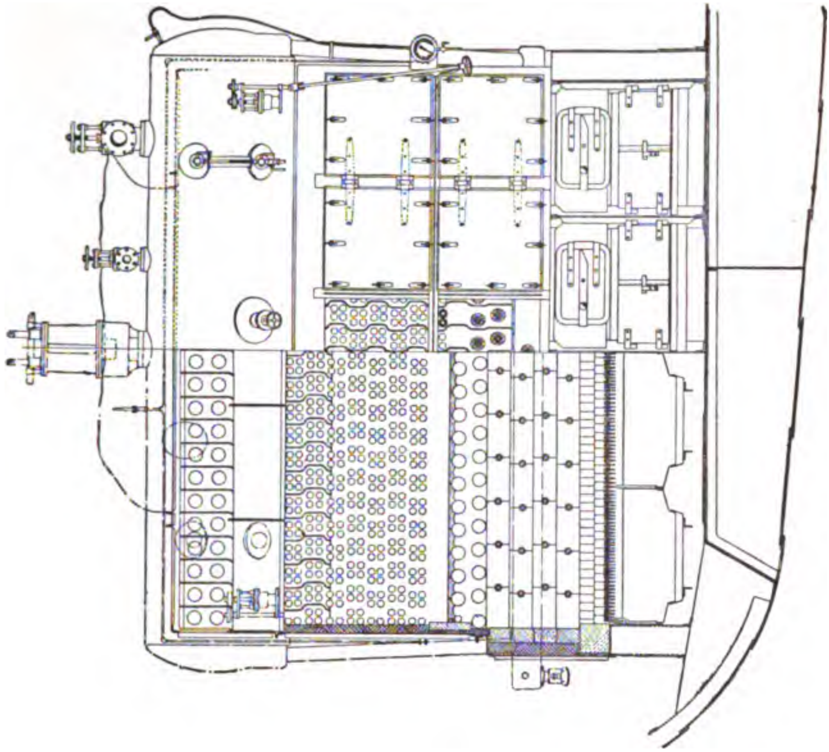


FIG. 58.

Durability.—As regards the durability of the Babcock boiler, many years' experience has now been obtained, and they give satisfaction in this respect. H.M.S. 'Queen,' after a two years' commission in the Mediterranean, returned home, and no expenditure was incurred at the dockyard to make good boiler defects, the vessel being re-commissioned again for a further period of service.

Comparison between Babcock and cylindrical boilers.—The report of the Admiralty Boiler Committee showed the Babcock and Wilcox boiler to be suitable for naval purposes, and the Committee obtained on their trials the high efficiency of 81 per cent. with the Babcock and Wilcox boilers of the 'Hermes.'

The Committee also carried out in the ss. 'Saxonia' a trial to



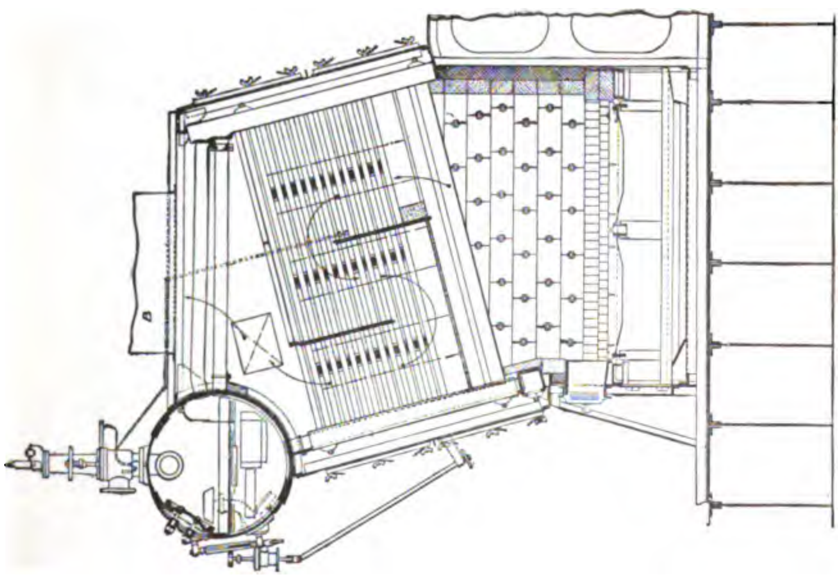


FIG. 59A. Babcock & Wilcox boiler—latest type. FIG. 59B.

show the capabilities of the cylindrical boiler in the mercantile marine, and fitted under the best conditions. The results obtained in these two ships, viz. the 'Saxonia' and the 'Hermes,' are of great interest, and they are compared in the table, p. 89.

In the case of the 'Saxonia' Howden's air-heating arrangement

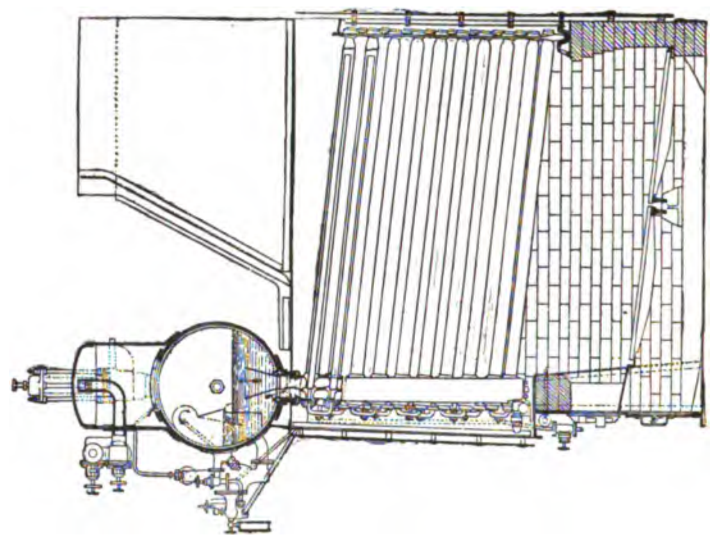


Fig. 61.

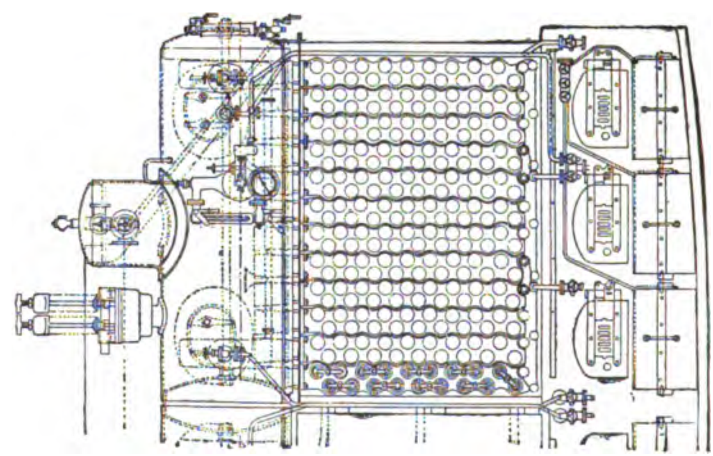


Fig. 60.

Niclausse boiler.

was fitted, and by this means an additional 4 per cent. of the heat of combustion was utilised. The 'Hermes' was not so fitted, although the air-heating system can also be used in association with the Babcock and Wilcox boiler. Therefore, in comparing with the 'Saxonia,' this has to be allowed for.

Niclausse boiler.—This is illustrated in Figs. 60 and 61, and has been fitted in a considerable number of vessels, especially in foreign

	'Saxonia'	'Hermes'
Transmission of heat units per sq. ft. of heating surface .	5,416	6,440
Lbs. of coal per sq. ft. of grate	20·6	20·0
Temperature of feed water F.	175°	88°
Pressure of steam in lbs. per sq. in.	199	227
Actual evaporation per lb. of coal	11·3	10·25
Equivalent evaporation per lb. of coal from and at 212° F.	11·33	12·17
Thermal efficiency per cent.	82·3	81·0
Actual evaporation per sq. ft. of heating surface in lbs. .	5·14	6·2
Equivalent evaporation per sq. ft. of heating surface from and at 212° F.	5·5	7·3
Temperature of funnel gases F.	396°	481°

navies. It consists of a series of slightly inclined double tubes, one inside the other, generally called 'Field' tubes, attached at the front end in such a manner that the colder water flows down the inside tube, and returns to the front between the two tubes when heated by the action of the fire and hot gases on the larger outside tube. Each vertical row of tubes is attached at the front to a separate square pipe or header, making the currents of each series quite independent of the others, and vertical diaphragms are fitted in these headers which completely separate the down-coming water at the front from the ascending currents of hot water and steam at the back as shown in section at the upper part of Fig. 61 and in Fig. 61A.

There are a series of such headers placed side by side, and they all lead into a top collector, fitted with a diaphragm and arrangements for keeping the entering feed-water and descending currents separate from the currents of hot water and steam ascending from the headers. The feed water is discharged into the steam space of the top collector, where it is raised in temperature by the steam, and an arrangement of screen plates and blow-off pipe enables any deposits to be blown off. The back ends of the tubes are supported by being allowed to rest in holes formed in a suitable plate or wall.

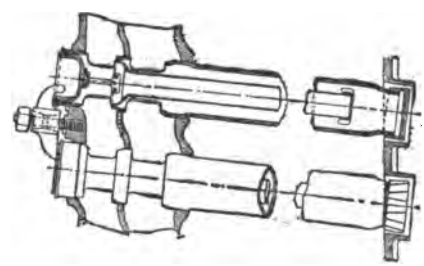
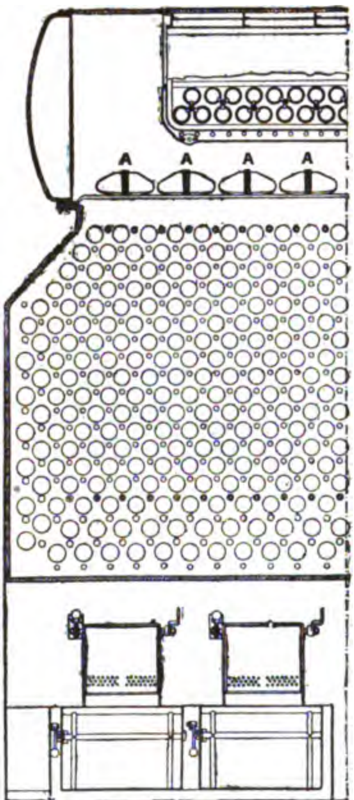


FIG. 61A.

The means of attachment of the tubes to the headers is by coned surfaces on the tubes, bearing on similar coned surfaces in the headers, with outside dogs and nuts as a safeguard in case of the displacement of a tube. A detailed view of the attachment of tube and header, also the back support, is shown in Fig. 61A. This joint appears to give no trouble by leakage, or in any other way, and when the water has been run down, a tube can be withdrawn for examination and replaced again in a few minutes. The headers are connected at the bottom by a blow-out pipe, but since each tube slopes downward with a closed bottom end, it cannot be emptied of water except by removing all the tubes, or by a pump and syphon. It is fitted in several war ships of the British, French and other navies.

Dürr boiler.—This boiler, shown in Figs. 62 and 63, has been extensively used in Germany, and is fitted in two British cruisers. It acts on the same general principle as the Niclausse boiler, i.e. it has



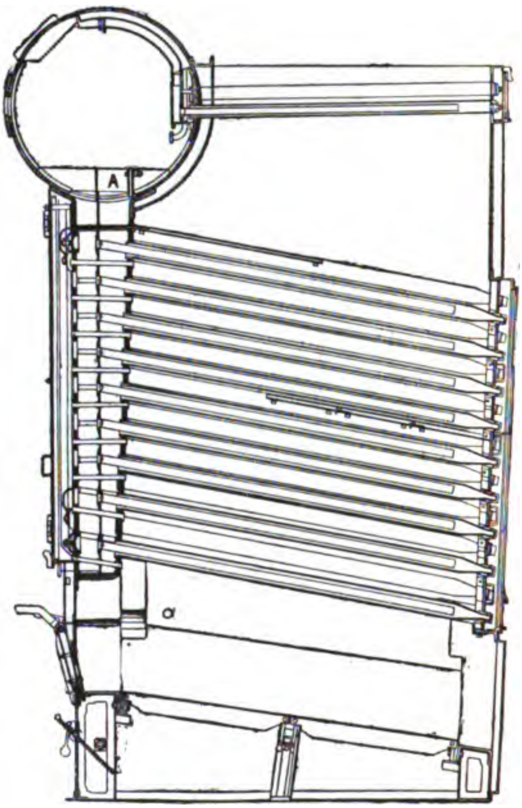


FIG. 62 Dürr boiler. FIG. 63

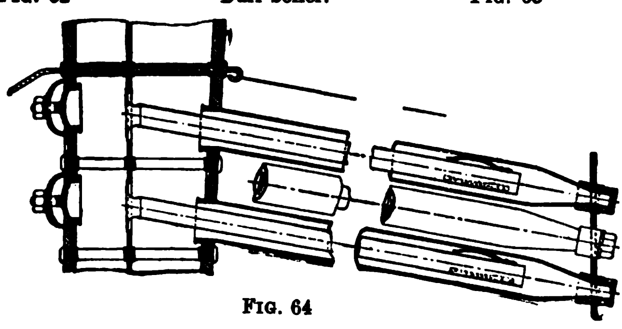


FIG. 64

slightly inclined 'Field' tubes, consisting of a small tube inside a larger one, the principal difference between the two types being that in the Dürr boiler the front ends of the tubes are attached to a continuous

water chamber formed of steel plates stayed together by a large number of short screwed stays, a continuous division plate being fitted to separate the downward and upward currents. The top of this water chamber is attached to and opens into a large steam drum running across the front of the boiler. A series of flat stays marked A being provided to allow for the weakening due to the plate removed. Water flows from the steam drum down the outside half of the water

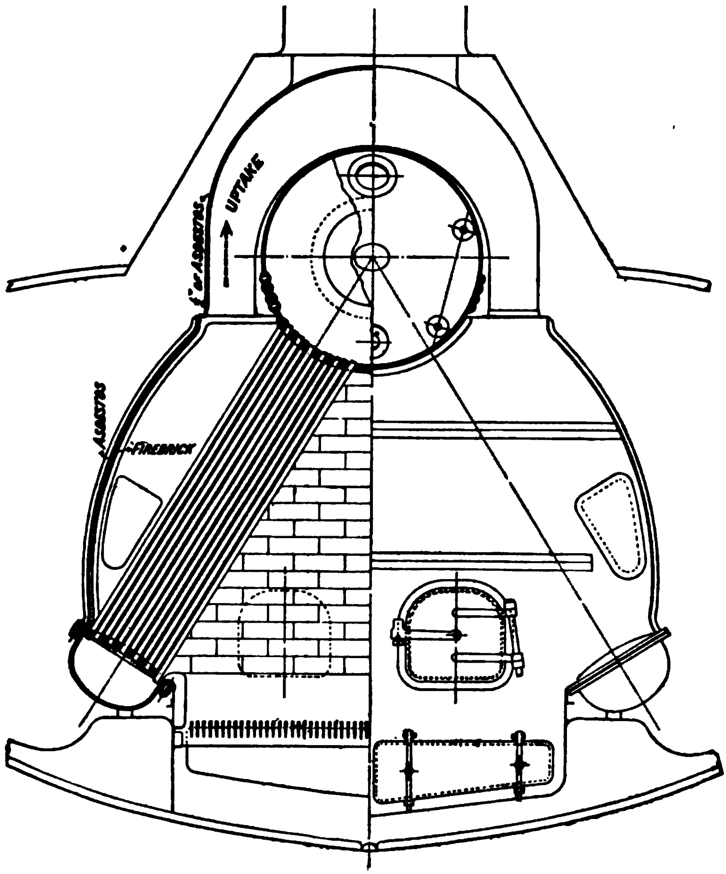


FIG. 65.—Yarrow boiler.

chamber, through the series of inner tubes, and returns to the front through the annular spaces between the two tubes. The outer tubes being exposed to the heat from the fire generate steam, which ascends with the water through the annular spaces to the inner half of the front chamber and thence to the steam drum. The steam disengaged from the water then passes through a few superheating or drying tubes at the top of the boiler of the same construction as the steam generating tubes, before proceeding to the engine. The outer tubes

are closed at the back ends by screwed gunmetal caps which may be removed for draining and cleaning purposes, space for access to the backs of the boilers for this purpose being allowed. Baffles are fitted to ensure that the gases traverse the bulk of the tube surface. Two of the rows of short screwed stays have a small hole drilled through them to allow of the insertion of wires shown in the sketches and attached to the baffles, which enables them to be shaken in order to detach any soot that may accumulate.

A sketch of the tube ends is shown in Fig. 64. The joint of the outer tube with the water chamber is made by the surface being coned and being simply pressed into a conical seating. The axis of the conical joint is at an angle with that of the body of the tube, owing to the plates of the water chamber being nearly vertical while the tubes are considerably inclined. It will be seen that the outer tubes are connected only to the back plate of the water chamber and the inner tubes are connected only to the middle diaphragm, whereas in the Niclausse boiler the outer tubes are carried to the front of the boiler, with suitable orifices for the entry and exit of steam and water, while the inner tubes are attached to the front plug.

Yarrow boiler.—The Yarrow boiler (Fig. 65), is fitted in a large number of British and foreign warships of all sizes from battle-ships down to torpedo-boats. In some cases it has been combined with a proportion of ordinary cylindrical boilers. As fitted in high-powered cruisers and battle-ships in the British navy, the size of the tube is larger than is the case in the installations in smaller ships. Depending on the size of boiler-tube the boiler is described as the 'Yarrow large tube' or 'Yarrow small tube' boiler, the former being fitted in the larger ships where the weight can be allowed and the latter in torpedo-boat destroyers and such boats where the weight has to be reduced to a minimum. The difference consists almost entirely in the size of tubes—the large tube type having tubes of $1\frac{3}{4}$ inches diameter and the small tube type from 1 to $1\frac{1}{4}$ inches, with a small increase of diameter in those next the fire. This boiler consists of a steam drum at the top, in the centre, and at the lower end on each side a small water chamber with nearly flat tube plates between which and the steam drum are a series of tubes which form the heating surface. The tubes all deliver their steam and water to the steam drum below the water line. The early boilers had external return water-tubes, but the latter are sometimes omitted, in which case the return of water must take place down those tubes which are exposed to the least intense heat. A few are now generally shielded from the fire at the ends by means of a diaphragm plate fitted between the tubes, to facilitate the return flow of water from the steam drum to the lower water chambers.

The boiler tubes which are straight are secured at the ends by being expanded by the roller expander. The general absence of curvature in the tubes is a distinguishing feature of the Yarrow boiler among those with comparatively small tubes. In boilers for the latest ships of the British navy the two fire rows of tubes on each side are curved slightly in order to prevent any tendency to leakage, which is greater in those rows than in the others farther removed from the fire.

In the latest variety a diaphragm is sometimes fitted along the lower water chambers to enable the feed water to pass up through the

rows of tubes farthest from the fire, by which an economy results similar to that with the economiser type of Belleville boiler.

The White-Forster boiler, illustrated in Figs. 66 and 67, is similar in general arrangement to the Yarrow boiler. There are two lower water drums connected by the generating tubes to an upper steam and water drum, the water level being arranged as to entirely drown all the tubes. The radius of curvature of each tube is the same, and this is only sufficient to determine the direction of movement due to expansion, but

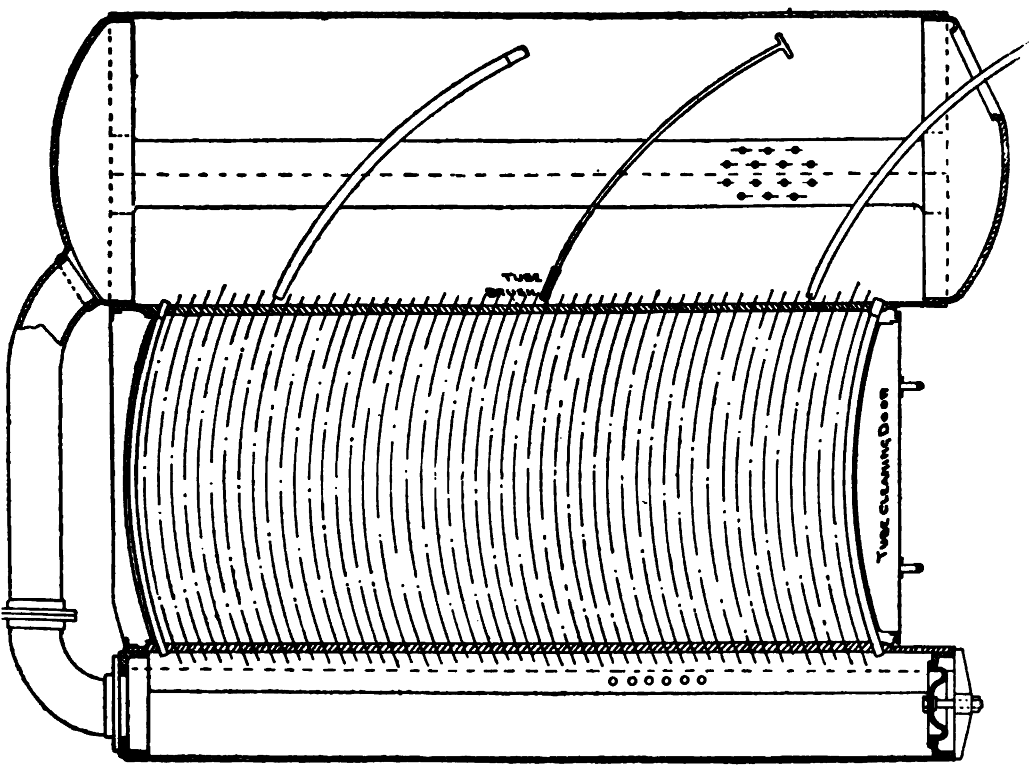


FIG. 66. —White Forster boiler.

it prevents possible tube destruction and leaky ends. It also facilitates cleaning and repairs, and reduces spare gear.

The tubes are arranged in position in a transverse section in a similar manner to the staves in the section of a barrel, and the continuation of the line of curvature of each tube passing through the top drum in line with the end manhole, allows each tube to be inserted or withdrawn through this manhole ; the withdrawal of any tube without disturbing the remaining tubes is thus facilitated, and as each tube has the same curvature they can be readily cleaned internally by a tube brush having a rigid handle curved to the same radius.

The boiler and its casings are constructed entirely of mild steel, no castings being employed, large downtake tubes are usually fitted at the back, and ordinary manhole doors are fitted to all drums.

As the inclination of the generating tubes is considerable and varies

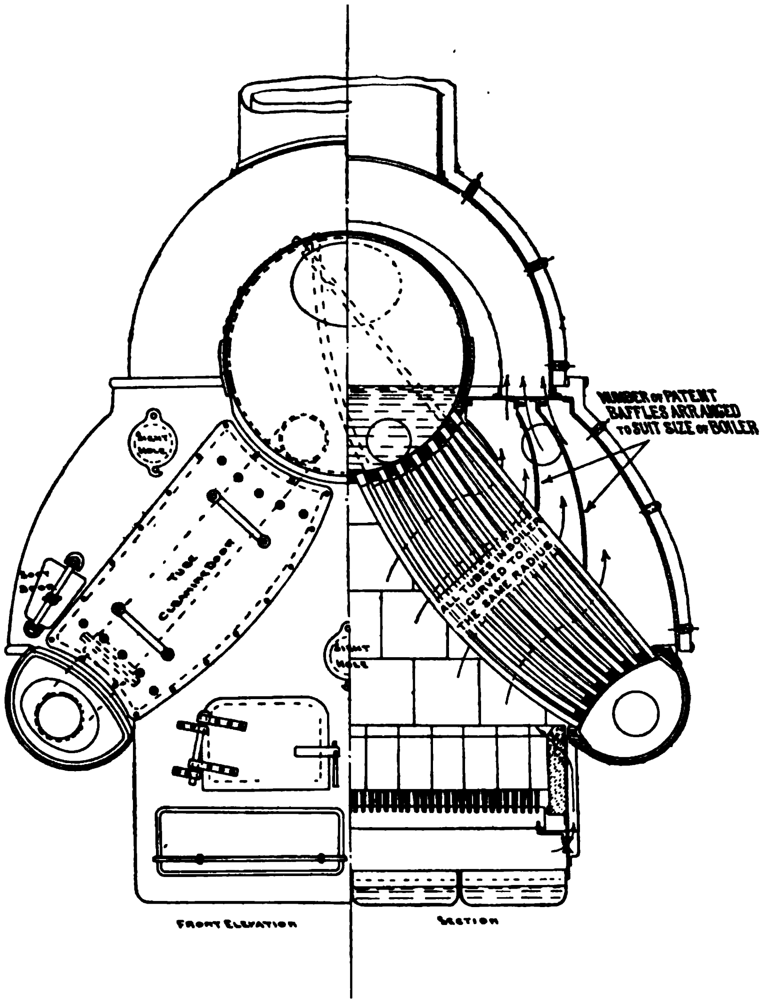


FIG. 67. —White Forster boiler.

between 40° and 60°, the circulation is restricted and rapid, keeping the tubes free from deposit. No special water baffles or separator plates are found necessary, only the usual internal steam-pipe being fitted.

Care is taken in designing the casings and uptakes to make the boiler as efficient as possible. No plates, tube grips or other baffles are fitted amongst the tubes, which would interfere with tube-sweeping,

but patent baffles are so fitted as to divide the uptake into two or more parts, in such a manner that the gases are drawn equally over the whole tube surface, and at right angles to it, the resistance and air-pressure being thus reduced to a minimum. Air holes are arranged at the sides and ends for the admission of air above the grate, and sight holes fitted for the inspection of the uptakes and combustion chamber.

Large doors are fitted at the front end, so that the whole length of tubes is exposed for cleaning, and a brush can be passed between the tubes from end to end of the boiler, so that the tubes are easily cleaned inside and out.

This type of boiler is simple, it possesses facilities for repair, and its efficiency compares very favourably with other types.

The Thornycroft boiler.—There is more than one variety of this boiler. The 'Speedy' type, Fig. 68, consists essentially of a central upper steam cylinder A, and two smaller lower water cylinders near the level of the fire bars. A series of steam generating tubes of small diameter are fitted between the upper cylinder and each of the lower water cylinders, and secured at each end by being simply rolled into the cylinder plating by means of the roller expander, the parts of the cylinders into which they are rolled being made thick enough for this purpose. These tubes form practically the whole of the heating surface of the boiler, and the inner row on each side is curved in such a manner that they are close together at the top and form the roof of the furnace or combustion chamber. Each of these two rows is made into a wall of tubes, through which the gases cannot penetrate except through spaces left at the bottom of the tubes. This is effected by bending the tubes just above the lower cylinder and fitting each alternate tube into the space between its neighbours, so that it forms a closed wall, except for the spaces E F at the bottom. The curvature of the tubes allows considerable freedom of expansion.

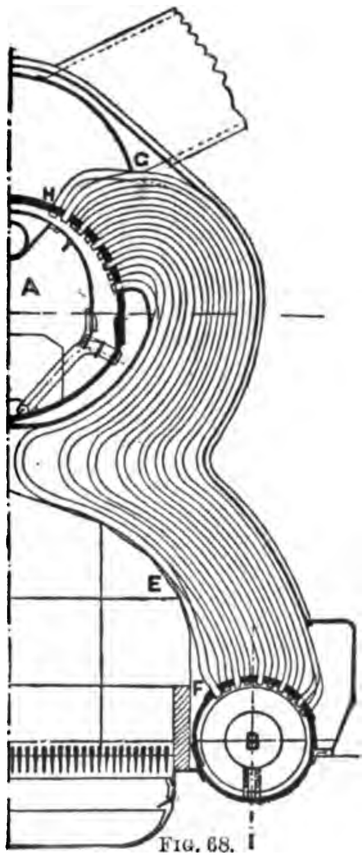


Fig. 68.
Thornycroft (Speedy) boiler.

The outer row on each side is similarly made into a wall of tubes through which the gases cannot escape except through openings G H left for this purpose at the top. The gases, therefore, generated in the furnace enter among the tubes through the opening left at the bottom of the inner walls of tubes; they ascend, traversing the whole of the

tube surface between the inner and outer walls, and emerge on their way to the funnel through the openings near the top, left in the outer wall of tubes. The whole of the generating tubes enter the top collector above the water level; there is therefore no possibility of any water being returned to the lower water chambers through these tubes, so that special return water tubes are essential. These return water tubes, two in number, are fitted at one end of the boiler; they are of large diameter, and connect the top cylinder with each of the lower water chambers outside the smoke casing.

The flame and gases around the tubes cause a rapid circulation of water upwards through the tubes, and some portion is evaporated into steam. The steam and water emerge at the tube ends against a baffle plate, which guides the water being discharged down to the bottom of the collector, and prevents the entry of spray into the internal steam pipe which runs along the cylinder at the top immediately below the baffle plate, to draw off the steam produced. A light steel casing is fitted over the tubes on the outside, and the ends of the boiler are formed by flat casings, and in the vicinity of the fire-bars a brick lining is fitted, forming a boundary for the fire. The feed-water is admitted near the centre of the top cylinder.

The trials of the 'Speedy' showed that these boilers could not be worked at high power with much salt water in them without considerable priming—in fact, this feature is common to all the various types of small-tube boilers. Special care is therefore necessary by frequent examinations to insure that the glands of the condenser tubes are properly packed and adjusted to prevent leakage.

Thornycroft (Daring) type.—The length of the 'Speedy's' boilers to allow for return tubes is considerable, the tubes are not very accessible, and the height of the furnace is not great, so that in order to reduce weight, also to give an improved furnace and greater means of access to the tubes, the Daring type of boiler was designed (Fig. 69).

This type obtains two furnaces in each boiler and a greater amount of fire-grate in the available space. It consists of an upper cylinder A, into which the upper ends of the generating tubes are rolled. Vertically below this is the principal lower cylinder B, to which the lower ends of the majority of the tubes are attached. Two smaller water cylinders, D, are arranged on each side, and the furnaces are situated between them. Three rows of generating tubes connect the upper cylinder to the water cylinders and form the outside boundaries of the furnaces, the inner boundaries being made by the tubes connecting the upper cylinder with the principal lower one. Of the three rows connected to each of the small water cylinders, the outside two touch each other, and so form a water wall which confines the flame and gases. The tubes are so curved as to leave a considerable space, C, between the two groups on each side; the gases are discharged into this space, which leads to the uptake.

The inner and outer rows of tubes of each centre group are formed as walls of tubes, except at the lower part of the furnace side and upper part of the uptake side, where spaces are left for the entry and exit of gases. The gases, after leaving the fire, enter through the spaces E, at the bottom of the outer row, pass up between the walls among the tubes,

emerge through the spaces F at the top of the inner rows to the space C, and proceed along this space to the end of the boiler and thence to the funnel.

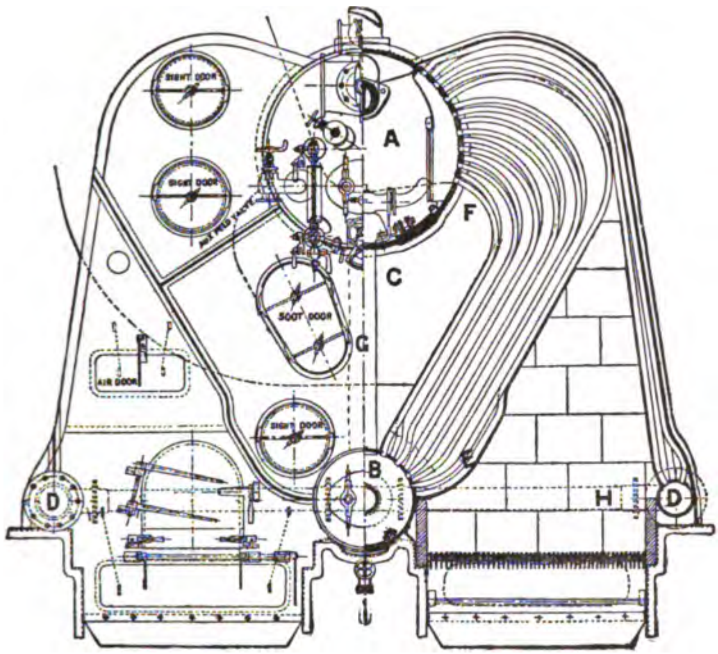


FIG. 69.—Thornycroft (Daring) boiler.

Provision is made for the return of water to the lower chamber by fitting a series of vertical return pipes G between the bottom of the upper chamber and the top of the lower cylinder. These return tubes therefore pass through the flue of the boiler and abstract a certain amount of heat from the gases before they escape from the boiler.

FIG. 70.

The small outside water chambers, D, are connected at the ends by means of horizontal pipes, H, with the central lower chamber, so that they obtain their share of the water brought down by the return pipes.

All the tubes in the Thornycroft boiler of both types illustrated discharge their water and steam above the water-line. Tubes so situated are spoken of as being 'not drowned,' to distinguish them from tubes with the upper ends below water or 'drowned'

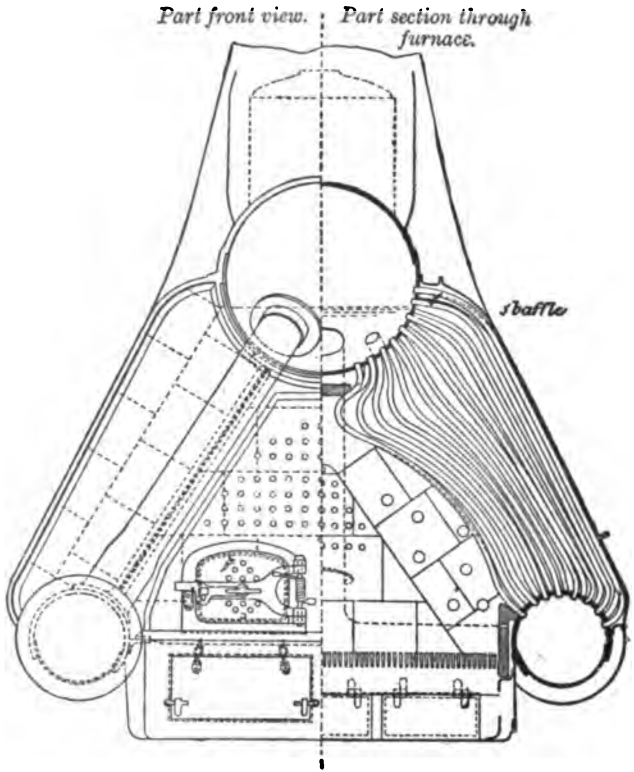


Fig. 71.—Normand boiler.

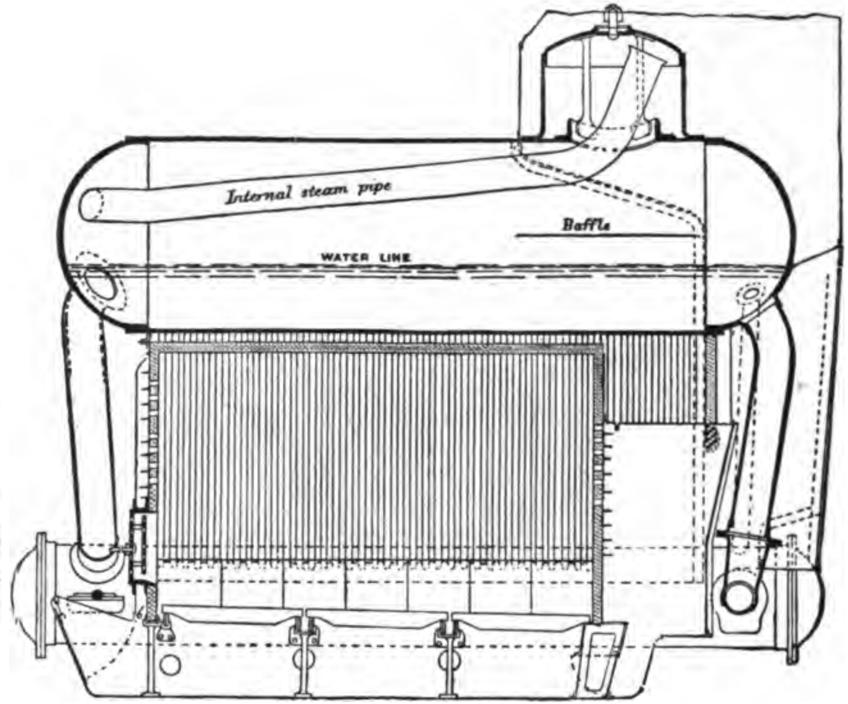


Fig. 72.—Normand boiler.

tubes. Certain advantages are claimed for this arrangement, principally that of steady circulation undisturbed by sudden collapse of steam bubbles on exit from the tube, or any tendency of water to return through the heating tubes. The tubes above the water line are, however, specially liable to corrosion, so that in the latest Thornycroft boilers the upper part has been rearranged and the tubes all discharge practically below the water line, as shown in Fig. 70, which represents the upper part of later boilers of this type, fitted in lieu of the arrangement of upper tubes shown in Fig. 69.

Normand boiler.—The Normand type, which with more or less modification has been fitted in a number of torpedo-boat destroyers, is shown in Figs. 71 and 72. There are the usual top cylinder with two lower water chambers. Large external return water tubes are also fitted at each end. The tubes enter the top chamber mostly below the water level, but a few of them discharge above this level. The two outer rows of tubes on each side are formed into a wall of tubes, as in the Thornycroft type. The furnace does not extend the full length of the boiler, and in its vicinity the tubes are arched upwards so as to leave space for a combustion chamber, whereas beyond the furnace, where there is no necessity for so arching them, they are of much less curvature, and as no combustion chamber is required at this part a larger number of tubes are fitted. It will be noticed that all the tubes adjacent to the fire have a considerable amount of curvature, while even at the ends beyond the furnace, where the heat is less intense, the tubes have also a certain amount of bending, which allows for expansion when heated.

The gases proceed from the fire among the tubes, as shown by the arrow on Fig. 73, which shows a half section through the tubes, and traverse the length of the boiler to the uptake end, where they pass below a brick deflecting plate to the space around those tubes that are less bent. They then rise on each side in the smoke casing, unite, and proceed to the funnel. A large steam dome is provided to which the steam pipe is led.

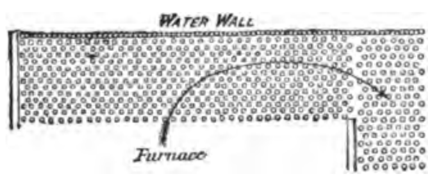


FIG. 73.

In this boiler provision is made for the admission of air above the fire by means of a small air casing at the front and back, and a series of small holes about one inch in diameter leading through the brick work to the space above the fire. This casing is connected with the air supply below the ashpit, and serves the double purpose of keeping the front and back of the boiler cool and sending a supply of warm air through the holes into the space among the gases above the fire. The series of steel pins project into this air space from the brick lining, and the air in passing these pins absorbs heat from them as well as from the surface of the steel casing. As the holes in the brickwork and steel casing must correspond to insure the passage of air in this manner, the bricks have to be well secured to the casing.

CHAPTER IX.

BOILER MOUNTINGS AND BOILER-ROOM FITTINGS.

Smoke-box.—The gases emerging from the boiler tubes are received by a structure of steel plates bolted to the end of the boiler for this purpose, and termed the smoke-box, in the front of which a series of hinged doors are fitted, so that by opening these doors the ends of the tubes are accessible for sweeping or repair. The smoke-box doors are generally on vertical hinges, and are fastened by two or more clips. The front of the smoke-box slopes away from the boiler as it proceeds upwards, so that the area increases with the volume of the products discharged from the tubes.

Funnel and uptake.—The hot gases after leaving the smoke-boxes are conveyed from the different boilers, through passages called the uptakes, as shown in sketches Figs. 77 and 78.

The uptakes are therefore the parts between the funnel and the smoke-boxes, and they are rectangular in section, and should be fitted with plates separating the uptakes of the several smoke-boxes, carried a sufficient height to cause the gases to be moving in approximately the same direction when they mingle together, so as to avoid loss from confusion of currents.

The smoke-boxes, uptakes, and funnel are surrounded by an outer casing, forming an air space which prevents the heat radiated from being excessive. This outer casing A is carried continuously from the lowest part of the uptakes to the top of the funnel, and is fitted with a hood a little distance from the top to prevent the entry of rain inside the casing, as shown at H in Fig. 77.

The funnel is carried between the various decks inside a hatch C fitted for this purpose, called the 'funnel hatch.' An additional air screen B is fitted to the uptakes and lower part of the funnel for carrying off the heated air from the top of the boiler room. This casing is carried a sufficient height above the upper deck to prevent inconvenience to people in the vicinity from the hot air discharged.

The area of the funnel in marine boilers is usually from one-seventh to one-eighth the area of the fire-grate.

Funnel dampers.—Hinged dampers are generally fitted in the uptakes of water-tank boilers to enable each boiler to be shut off when not at work, and also for use when cleaning fires. In the example illustrated there is one combustion chamber to each two furnaces, so that one damper is fitted to the uptake space from each two furnaces. These dampers are generally fitted so that there are no means of closing them permanently, and if released they should move to the open position. These dampers are shown at D D in the uptake section, Fig. 77.

Telescopic funnels.—Many of the old masted ships of the Royal Navy were fitted with telescopic funnels, so that when proceeding under sail alone they could be lowered to clear the space necessary for working the sails, but this fitting is now practically obsolete.

Funnel stays.—The funnels are stayed by wire ropes carried from the top of the funnel to the ship's sides, usually called funnel stays or guys. These are fitted with adjusting screws to regulate the strains, and should be slackened before raising steam to allow for the expansion of the funnel as it becomes heated. In modern vessels, funnels of from 90 to 100 feet high from the furnaces are common, and in such cases additional stays are fitted to the outer casing about 15 feet below the top of the funnel as an additional precaution when rolling. Gantling blocks are fitted at the funnel top with chains to facilitate painting.

Funnel cover.—When in harbour, or any funnels are not being used, portable covers are fitted to the funnel to prevent rain water coming down and corroding the uptakes, &c. The covers are kept a little above the top, to allow space for the escape of the smoke from the small fires used for airing and warming the boilers. The covers are now usually of canvas, which rests on angle supports, and a ladder-way inside facilitates their removal and replacement. In the earlier ships the covers were of steel, and small derricks were fitted at the funnel top to facilitate handling them.

Stokehold ventilation.—Care is required to ensure the proper ventilation of the stokehold. When natural draught only is used, screens are generally required to keep the downward current of cool air separate from the upward current of warm or vitiated air, otherwise these currents may be attempting to move in different directions at the same place and so destroy the circulation. Not only has the supply of a sufficient quantity of air for the fires to be arranged, but the health and comfort of the men working in the stokehold have to be provided for by the removal of the hot air, and provision of cool fresh air for respiration and reduction of the stokehold temperature.

In the Royal Navy blowing fans, as described in Chapter V., are now always supplied for the stokeholds, even when forced draught is not used. This makes the supply of air to the fires less dependent on the funnel in case of injury to the latter, simplifies the stokehold ventilation, and enables an ample supply of fresh air to be obtained under all circumstances. In the mercantile marine fans are also often fitted.

Figs. 77 and 78 show the complete stokehold ventilating arrangements of a war ship fitted for forced draught; the various funnel and other screens and casings are indicated, and arrows show the direction of the upcast and downcast currents of air.

Ash tube and engine.—For raising ashes to the deck to be thrown overboard, at least one ash-tube is fitted in all steam vessels, except the smallest, leading from stokehold to deck. These tubes are often used as ventilators. The stokehold ends are either carried low down permanently, or made telescopic so that they may be lowered when raising ashes, to prevent accident to persons who might otherwise get beneath them.

With closed stokeholds, special arrangements are necessary to enable ashes to be raised when under air pressure without considerable losses of air. To effect this the ash-buckets are sometimes carried up and down in a cage which works practically air-tight in the tube; but a more simple and equally efficient plan is shown in Figs. 79 and 81. It consists of a block A, of sufficient weight to resist the air pressure, which is carried loosely on the chain and acts as a valve at the top of the shoot when the bucket is being lowered, enabling the door B at the bottom to be opened without loss of air.

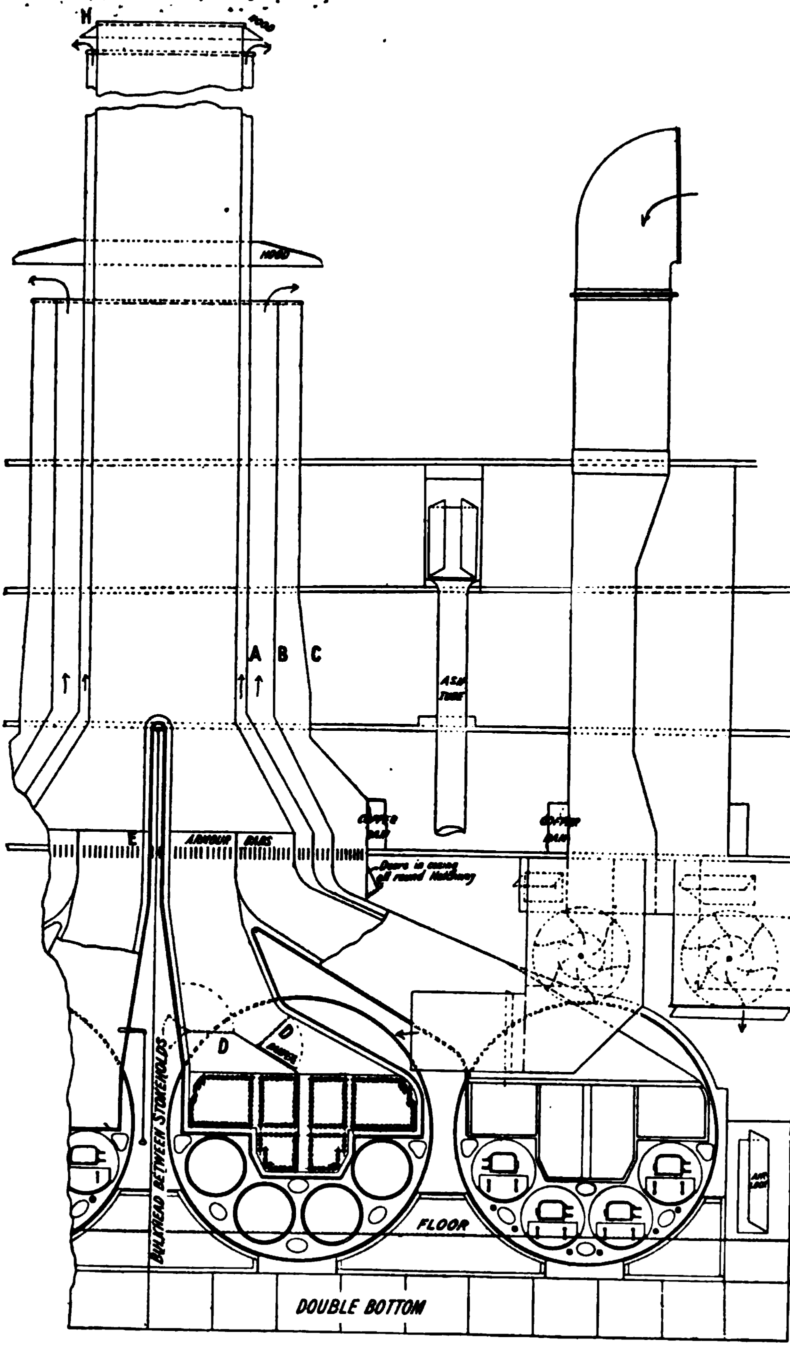

H
HOOD
A
B
C
E
ARMOUR BARS
Doors in casing all round Hatchway
D
D
BULKHEAD BETWEEN STOKEHOLDS
FLOOR
DOUBLE BOTTOM

FIG. 77.

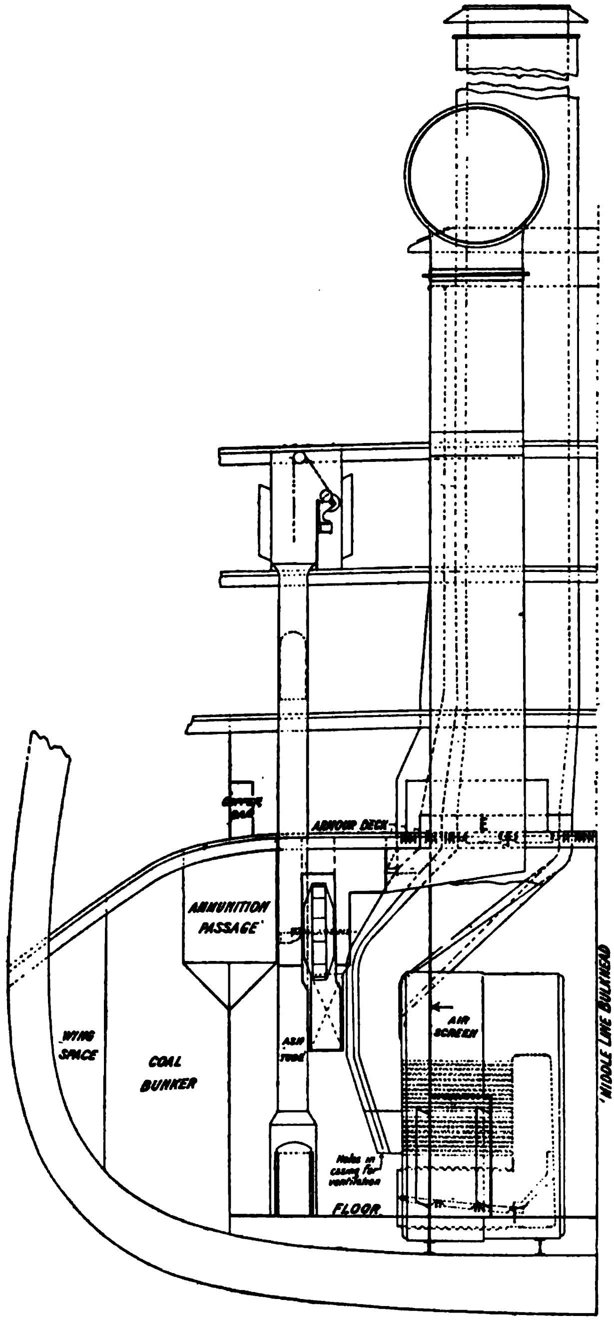

ARMOUR DECK
AMMUNITION PASSAGE
WING SPACE
COAL BUNKER
ASH TUBE
AIR SCREEN
Holes in casing for ventilation
FLOOR
MIDDLE LINE BULKHEAD

Fig. 78.

A small steam-engine is provided for raising the ashes, automatic gear being often provided, so that the engine stops when the ash-bucket is at the proper height. This gear often consists of a small ball on the chain, which in a certain position moves a fork attached to the steam valve of the engine by levers, and closes it. A gong, voice pipe, or other means of signalling from stokehold to deck is also fitted.

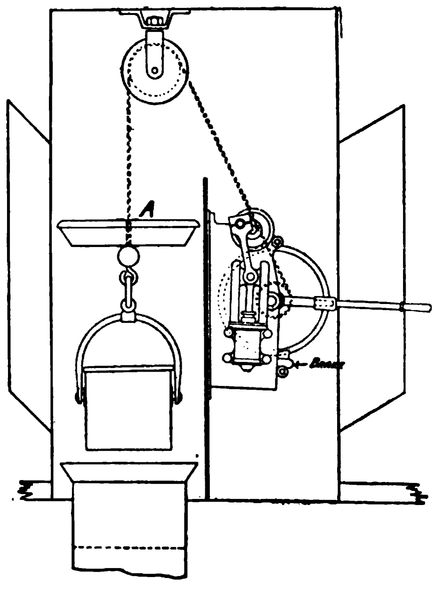


FIG. 79.

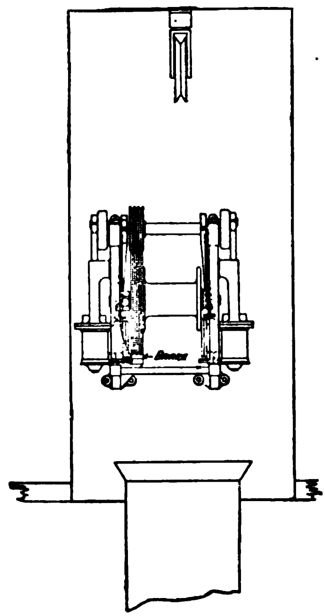


FIG. 80.

Figs. 79 and 80 show the arrangement of an ash-hoisting engine. In this example the engine works the chain-barrel by means of friction gear. The axis of the chain-barrel is carried in eccentric bearings, so that motion of the handle releases the chain-barrel from the engine and enables the bucket to lower itself. A further motion of the handle brings a brake into action which stops the bucket when required.

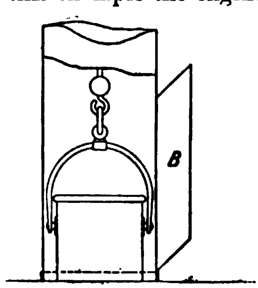


FIG. 81.

Armour gratings.—In all war vessels with protective decks, the machinery hatches, such as funnel hatch, the funnel itself, the air down-takes, engine room hatches, &c., are fitted with deep *armour bars*, as at E in Figs. 77 and 78, to prevent damage to parts below from shot or shell, and the newer vessels are provided with steel wire *splinter nets* a short distance below the bars to stop the smaller pieces of *débris*. The weight of the funnel and its casings and armour gratings is carried by the hatchway and deck at E through deep girders, so that the uptakes and boiler are relieved of this strain.

Arrangement of boiler mountings.—The usual position and arrangement of the principal mountings on a cylindrical marine boiler are

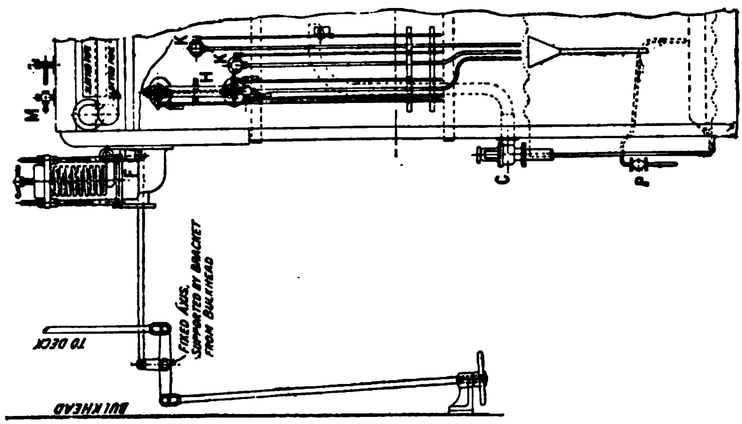


FIG. 84.

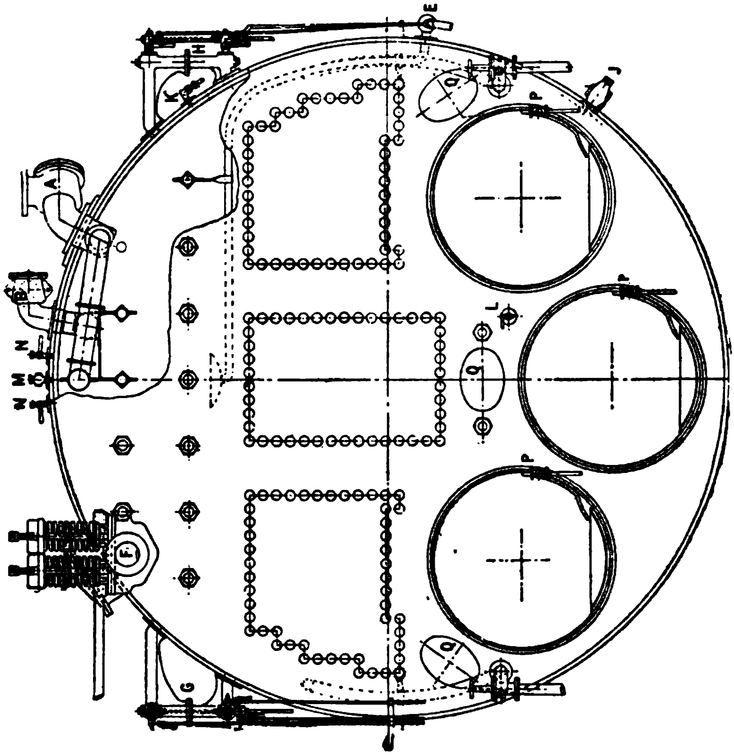


FIG. 83.

shown in Figs. 83 and 84. A and B are respectively the main and auxiliary steam stop-valves, C and D are the main and auxiliary feed-

valves, the main feed being on the right-hand side ; E the surface blow-off, with internal pipe, as shown, leading to a trough near the surface ; F are the safety valves, with rod for raising valves by hand, and behind the safety valves is the top manhole, through which access to the interior of the boiler is obtained, the lower manholes being indicated at Q ; G and H are the two stand-pipes, on which are fitted the glass water-gauges ; J the valve with hose connection for running off water ; K are the test cocks ; L the hydrometer cock ; M is an air cock on top of the boiler shell ; N cocks leading to the boiler pressure gauges ; while P P are cocks leading water to the ashpans, which are, however, not fitted in recent ships. Fig. 84 shows the independent lifting gear, enabling the safety valves to be raised either from deck or stokehold. This view shows rods for working the gauge and test cocks, which otherwise cannot be reached from the stokehold floor, and the lead of internal feed-pipes. In both views a portion of the top of the boiler is removed to show the internal steam pipes and the method of supporting them.

All valves, cocks, and other boiler mountings are made with spigots passing into the plates to prevent corrosion.

Safety valves.—If all exit from a boiler were closed, and heat continuously applied, the pressure would continue to increase until at length the boiler must of necessity explode. This is prevented by safety valves, which are designed and fitted so that when the pressure in the boiler exceeds the safe working pressure, they open and let the excess steam pass off into the atmosphere, and thus prevent danger.

Sketches of safety valves loaded with springs are shown in Figs. 85 and 86. The safety-valve boxes are fitted to suitable orifices at or near the top, and directly on the boiler, and distinct from the stop-valve box, internal steam pipe, or any other possible obstruction, and the valves are kept on their seats by springs of sufficient force to just resist the maximum working pressure. When the steam pressure exceeds this, it opens the valves and the steam escapes to the atmosphere through an orifice in the box, to which the waste steam pipe leading to the atmosphere is connected. The springs are placed outside the box to prevent corrosion.

Weighted valves were used with the first safety valves fitted. They were loaded by lead weights placed directly on the spindle above the valve. These valves for marine boilers had many disadvantages. The oscillation of the weights due to the rolling of the ship caused the valves and seats to grind away and become leaky. The reduction of the direct load on the valve by the heeling of the ship also caused waste of steam, and there were other disadvantages. They had, however, the advantage that when they began to open they did not require any increase of pressure to open them still further, as is the case with spring-loaded valves.

The more a spring is compressed the greater is the pressure required to compress it still further ; and, within the limit of elasticity of the spring, the pressure required to compress it a certain distance increases directly as the amount of compression. By employing springs of ample length and diameter so as to obtain sufficient flexibility no difficulty is experienced from this cause, so that spring-loaded safety valves are in general use for marine purposes.

The safety valves should always be fitted in a vertical position or

nearly so, wherever this can be arranged. It is undesirable to place them horizontally. The guide feathers or spindle on the valve are always made with a certain amount of clearance to prevent their sticking, and horizontal valves tend to drop from the central position, so as not to find their true seating, and so allow steam to leak past them.

The breadth of the face need not be more than from one-twelfth to one-eighth of an inch. With conical valves the seatings should be narrow, and fitted so as to be quite tight at the bottom of the cone; otherwise the actual area on which the steam pressure acts will be greater than the nominal area of the valve, so that steam will be

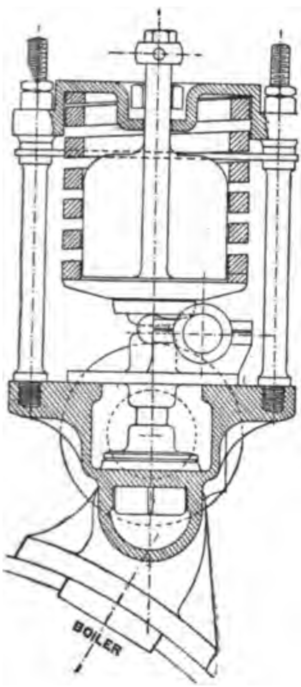


Fig. 85.

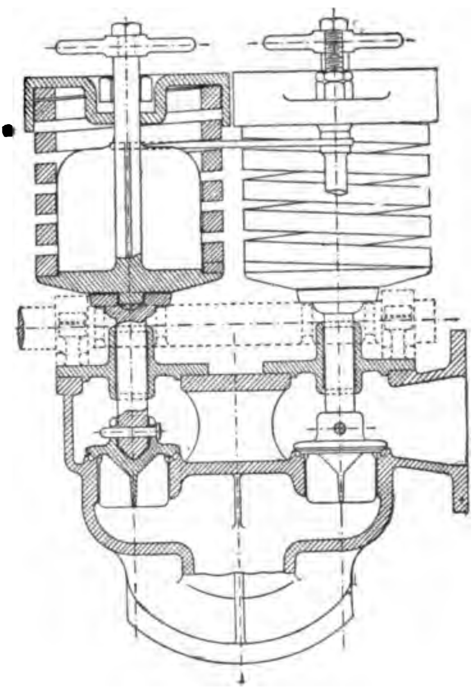


Fig. 86.

wasted, as the valves will lift before the steam attains the proper working pressure.

Area of safety valves.—It has been proved by experiment that when the absolute pressure of the medium into which the steam enters on passing the orifice is not more than one-half the initial absolute pressure of the steam, the velocity of issue, and therefore the weight of steam discharged per minute, is constant, so that the moderate pressure in the waste steam pipe when blowing off may be neglected in calculating the area of valves.

The requisite safety-valve area for each boiler should be divided between two or more valves, and not concentrated in one. The area for discharge of the steam depends on its periphery and lift, so that

when two or more valves are used, the lift required to liberate the steam is reduced because the periphery is increased. The division of the safety-valve area has also the advantage of reducing the danger from the valves sticking, as the probabilities are against all the valves becoming inoperative at the same time; and if one become set fast, the others would still act to free the boiler of undue pressure, when necessary.

The total area allowed for safety valves for the higher pressures may be reduced, because the rate of efflux of steam increases with the pressure, so that the safety-valve area should depend on the pressure of steam.

The area of the safety valves is often calculated from the grate surface only, but it should really be based on the *quantity of steam* the boiler is capable of producing when worked at full power, and not on the grate area, because the rate of combustion varies considerably under different circumstances. This is rendered especially necessary by the introduction of accelerated draught, which has largely increased the generative powers of boilers. The steam-producing power of the boilers may be represented approximately by the maximum I.H.P. developed, and we have seen that the rate of efflux of the steam will vary as the absolute pressure. If P represents the *absolute* working pressure of steam, the total area of safety valves required may be calculated from the formula

$$\text{area} = 3\frac{\text{I.H.P.}}{\text{P.}}$$

This formula is obtained by taking the rate of flow of the steam through the orifice, in pounds per minute, to be three-fourths the absolute pressure in pounds, and assuming the valve to lift one twenty-fourth of its diameter when blowing off the steam necessary to give the designed horse-power.

Accumulation of pressure.—The face of a safety valve should be so arranged in relation to the lip or body of the valve, that by the reaction of the escaping steam the pressure may be kept up when the valve lifts. The amount of accumulation of pressure when blowing off depends very much on small features of design in the safety valves, and by utilising the reaction of the escaping steam on projecting lips or by contracted orifices just beyond the valve seat, the accumulation may be reduced to very small dimensions, as the area on which the pressure acts is increased when the valves commence to blow. The object is to make the safety valve lift a considerable amount when it commences to blow, and to ensure its closing when the pressure has fallen about 4 or 5 lbs. below the safety-valve load. The sketches, Figs. 87 to 89, show three forms of safety valve with the arrangement for adding to the lift when it commences to open.

Safety valves of water-tank boilers are required by the Board of Trade to be tested with the boiler under full steam and full firing for about twenty minutes, with feed- and stop-valves shut off, and in this case the accumulation of pressure should not in any case exceed 10 per cent. of the loaded pressure. In the most recent Admiralty vessels this excess is limited to 7 per cent. with the stop-valve shut, but feeding the boiler with water in the usual manner.

Safety-valve springs.—In the Admiralty service all safety-valve springs have to undergo tests as to their strength and elasticity. The compression with the working load is first noted, and the spring is then further compressed through a distance equal to one-quarter the diameter of the valve, and if on the removal of the load the spring does not regain its original length it is rejected. The Admiralty practice is to make the amount of compression to give the working load equal to the diameter of the valve.

Lift of safety valves. Lifting gear.—Gear is always fitted so that the safety valves may be lifted by hand, and the amount of hand-lifting allowed is one-fourth the diameter of the valve. When this lift is attained stops come into operation to prevent further lifting. It must not be thought, however, when at work and blowing off even large volumes of steam from the boilers that the valves lift to this extent.

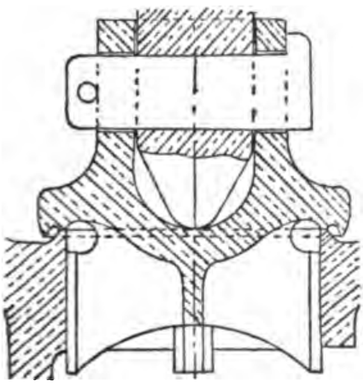


FIG. 87.

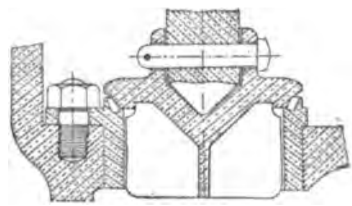


FIG. 88.

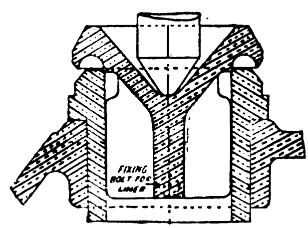


FIG. 89.

The actual lift when blowing off the full quantity of steam does not exceed about one twenty-fourth the diameter of the valve, or with the usual dimensions about one-eighth of an inch, so that the actual increase of load due to the extra compression of the springs should be small.

A washer or ferrule should be placed under the compressing screw, so that the spring cannot be compressed more than sufficient to give the maximum working load on the boilers, also a nut on the valve spindle, which prevents the extension of the springs beyond a small amount when being removed for examination, and so facilitates this operation.

The gear used for lifting the safety valves is shown in Fig. 84. It usually consists of levers acting either under the valves themselves or under collars on the valve spindles, these levers being worked by screw gear from the boiler room, and in the Royal Navy generally from the deck as well. The two sets of gear work independently of each other, and are so fitted that the valves may be lifted from either position without moving the gear at the other position. It is also arranged

that neither lifting gear impedes the automatic action of the valve in any way when acted on by the steam pressure in the boiler, as shown in the figure. All the joints in the safety valve easing gear should either be fitted with gun-metal bushes, or the joint pins should be of gun-metal, to prevent the gear rusting and setting fast. When separate seatings fitted into the valve chest are used, they should be so arranged that they cannot lift with the valve, as in this case there would be no outlet for steam. To keep the seatings from working loose, they are carefully secured by bolts or pins, as shown in Figs. 88 and 89.

Pressure gauges.—The steam pressure in the boiler—or, more strictly speaking, the excess of the pressure in the boiler above that of the atmosphere—is usually indicated by Bourdon's pressure gauges.

Fig. 90 shows their general construction and arrangement. A is attached to a cock on a small pipe connected to the boiler with a cock N (Fig. 83) at the junction. B B is a curved metallic tube, of elliptical section, as shown to enlarged scale at C, Fig. 91, which is closed at one end and open to the steam pressure through A at the other. The greatest breadth of the section of the tube is perpendicular to the plane in which the tube is curved. The closed end of the tube is connected by a sector D to a small pinion on the axis of the index finger E F, which points to a graduated arc. When the pressure inside the tube is greater than the external pressure, the tube straightens itself, and this causes the sector to act on the pointer and indicate the pressure.

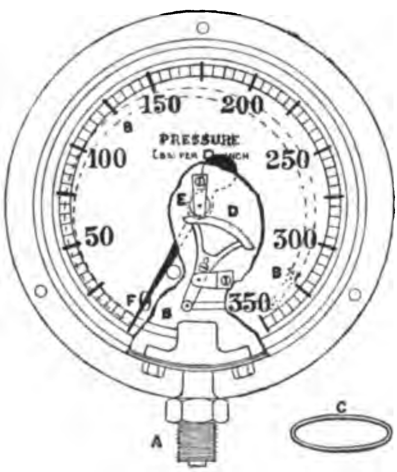


FIG. 90. FIG. 91.

The graduation of these gauges is obtained by comparison with a mercurial gauge, into which mercury can be pumped to sufficient heights to obtain the necessary indications on the Bourdon gauge.

In the Royal Navy each boiler is always fitted with two pressure gauges, in order to provide for the case of one gauge getting out of order, and to be a check on each other. With double-ended boilers two pressure gauges are fitted at the working end of the boiler—i.e. at the end on which the feed-valves are placed and where the person in charge would generally be—and a third pressure gauge is fitted at the other end. One gauge on each boiler is graduated to rather more than the hydraulic test pressure, to enable it to be used when testing the boiler at regular stated intervals.

Water gauges.—The level of the water in the boiler is indicated by a glass tube fitted between two asbestos packed cocks, one in connection with the steam space and the other with the water space of the boiler, while a drain cock and pipe are supplied for the bottom of the glass to enable it to be blown through and cleared. The general

arrangement is shown in Fig. 92. In most cases in the mercantile marine these gauge cocks are fitted on a brass pipe, known as a steady or stand pipe, the top of which is connected to the steam chest, the bottom being connected by an outside pipe to the lower part of the boiler, where there is but little disturbance of the water. Cocks are fitted at the top and bottom, where these pipes join the boiler shell (see Figs. 94 and 95).

Sometimes, however, the gauge cocks are bolted direct to the fronts of the boilers themselves, which is the usual plan in the Navy in the case of low boilers, in which the front of the boiler is clear of the uptakes, and room can be found. The only difficulties met with in this position arise from the tendency of any grease or light impurities floating at the water line, to enter the glass and dirty it, and when such boilers are forced much, unsteady indications are sometimes obtained, due to violent ebullition near the gauge orifice. With large return tube boilers, where the uptakes cover most of the front of the boilers, so that the glasses have to be fitted on the circular shell, the usual practice in the Royal Navy is to fit the gauge glass cocks on a brass casting, forming a short steady pipe, having a hole of comparatively large diameter (two inches), so that there is little or no danger of its becoming choked. This forms a convenient method of attachment to the shell, and the intervention of the stand pipe causes the indications to be comparatively free from disturbance due to rapid ebullition at the water surface. A sketch of this attachment is given in Fig. 93. On this system no cocks are required in the steady pipe, which is an advantage, as these may, by being inadvertently closed, be a source of danger. If the lower cock is bolted directly to the boiler shell and an internal pipe led down into the water space, a branch from this should be carried up open ended into the steam space as well as down into the lower part of the boiler, otherwise the indications will be unreliable; but in general it is better to dispense with internal pipes.

A small screw plug is fitted to each of the two cocks, opposite the hole making connection with the boiler or stand pipe, to enable a wire to be passed in to clear them when necessary. Two sets of water gauges are fitted to each boiler in ships of the Royal Navy; while in double-ended boilers, three are fitted, two at the working end and one at the other end.

The general practice in water-tank boilers is to fix the hole in the lower gauge cock at a small distance above the level of the highest part of the heating surface, so that when the water is just disappearing from the glass its level is from three to four inches above the highest part of the heating surface. The total length of the gauge glasses used in H.M. service for large boilers is 14, 16, or 18 inches, external diameter $\frac{5}{8}$ inch, and thickness $\frac{1}{8}$ inch. Allowing for the part of the length of the glass tube in the glands of the cocks, the depth of the water over the highest part of the heating surface when the glass is half full will be found to be 10 to 11 inches. For small boilers, as for pinnaces, cutters, &c., the gauge glasses are only $\frac{7}{16}$ to $\frac{1}{2}$ inch in diameter and $\frac{3}{32}$ inch thick, and shorter than the preceding lengths. The water side of the gauge mounting contains a small ball which, if the glass breaks, is forced up by the escaping water and stops the orifice, thus greatly reducing the amount of water escaping. In the

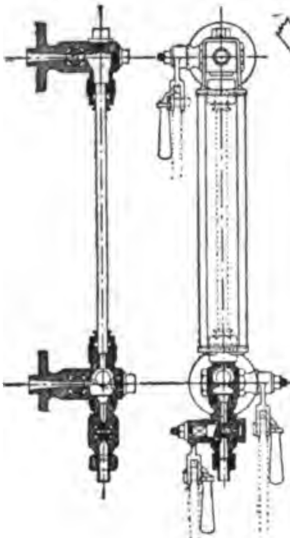


Fig. 92.

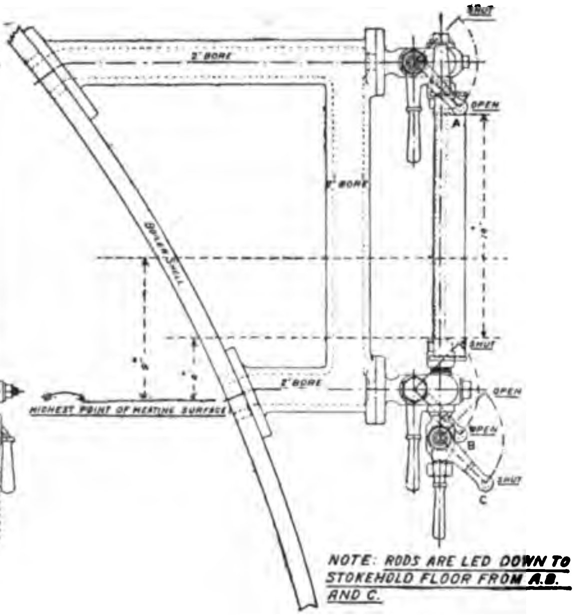


Fig. 93.

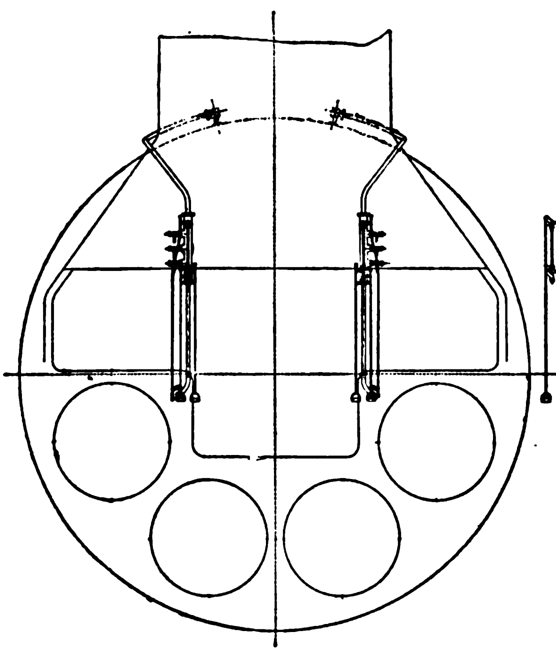


Fig. 94.

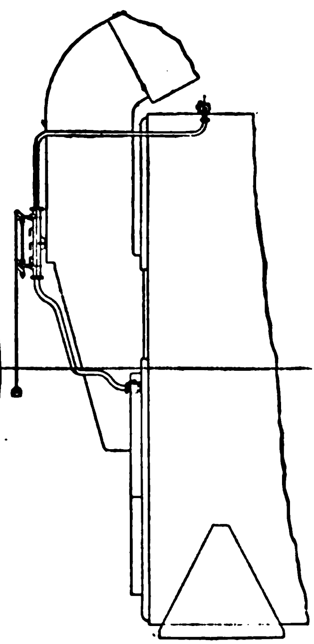


Fig. 95.

mercantile marine the glasses are larger in diameter, usually $\frac{3}{4}$ inch. Klinger's water gauges are described at the end of this chapter.

Effect of list on the boilers.—When the vessel has a list the relation between the level of the water in the glass gauge and the amount of water above the highest part of the heating surface undergoes an important and considerable change, and one that should be carefully ascertained in every steam vessel. The effect is to bring part of the heating surface much nearer the water line, and if the list is of any considerable amount, in a large boiler the heating surface will be left bare of water even when the gauge glass may be full. Fig. 96 shows a boiler placed with tubes fore and aft in the ship, and the lines drawn indicate that with a list of 7° even when one of the glasses is full, the wing combustion chamber is uncovered. Evidently in this case the other glass only should be worked with.

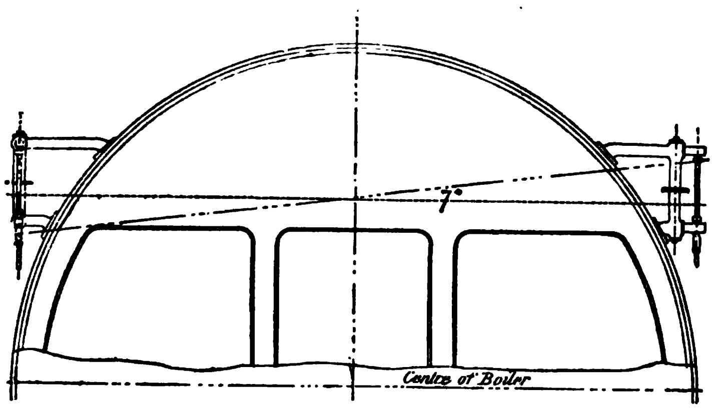


FIG. 96.

Fig. 97 shows a boiler with tubes placed athwartship. In this case the combustion chamber becomes uncovered with a list of $9\frac{1}{2}$°, even when the gauge glasses are full of water, although the combustion chamber is sloped away to reduce this angle. In this case, as both gauge glasses are affected alike, the boiler becomes practically unworkable. These sketches show actual examples in naval vessels.

Great importance is therefore attached to a proper understanding of the procedure necessary for working the boilers in the event of a permanent list, especially in naval vessels, where it may occur owing to a compartment on one side being flooded during an action. The best course to pursue should be considered by the engineers of each vessel.

Many water-tube boilers also are affected by a list in the vessel, the Belleville and Niclausse for example. In the former, when the tubes are placed athwartship, as the inclination of the tubes to the horizontal is only $2\frac{1}{2}$°, a list interferes with the circulation of water and a wedge is supplied for use under these circumstances, so as to keep the automatic feed-valve always open a minimum amount to insure a sufficiency of water. When the tubes are placed fore and aft, as in the great majority of British warships, this is unnecessary. In the Niclausse

boiler also, a list in one direction interferes with the flow of water down the internal supply tube of the boiler when placed athwartships.

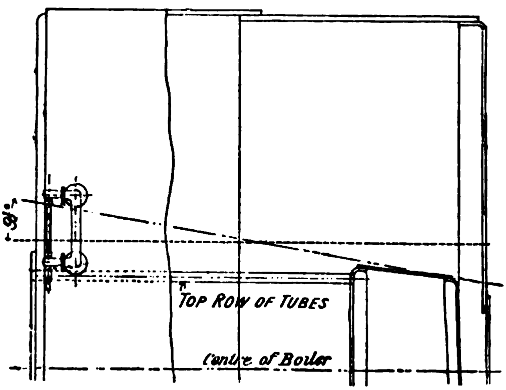


FIG. 97.

Test cocks.—On the front or shell of the boiler, small cocks, called test cocks, generally two in number, are usually fitted, and shown at K in Figs. 83 and 84. The orifice of the lower of these cocks is about two inches above the highest part of the heating surface, and that of the upper cock twelve inches above the highest part of the heating surface. The use of these cocks is to enable the level of the water to be ascertained approximately in case of accident to or derangement of the glass gauges. When working by these cocks it is clear that if, on

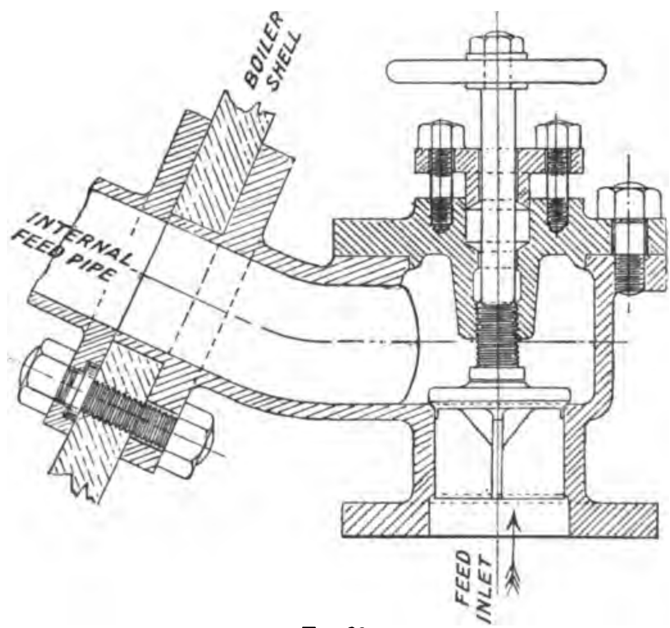
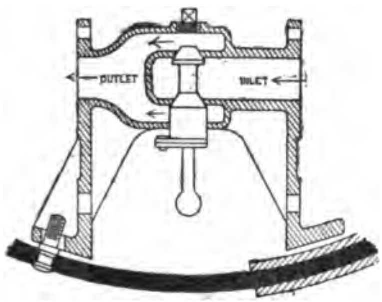


FIG. 98.

opening the upper cock, steam only issues, the water is not too high in the boiler; and as long as water can be drawn from the lower cock the

water-level is not dangerously low. These test cocks should evidently be fitted without internal pipes.

Feed-valves.—These are the valves through which the feed-water is admitted to the boilers. They are screw-down non-return valves, which may be kept closed, or their lift regulated, as shown in Fig. 98. There is no connection between the valve and the screwed spindle, the latter simply limiting the lift of the valve or pressing against it when closed. These valves are made non-return, so that in case of the feed-pumps ceasing action from accident or other cause, the water in the boiler may not be forced back through the pipes by the steam pressure. An additional valve is fitted in the delivery pipe of each feed-pump for completely shutting off the pump from the feed-pipes and so enabling any examinations and adjustments to be made.



ENLARGED VIEW OF VALVE

FIG. 99.

Two feed-valves are generally fitted to each boiler, one, called the main feed-valve, being in connection with the main feed-pumps, and the other, called the auxiliary feed-valve, connected with the auxiliary or donkey feed-pumps. In the Royal Navy the main feed-valve is placed on the right-hand side of the boiler, while the auxiliary feed-valve is placed on the left.

Automatic feed apparatus.—The feed-regulating apparatus for

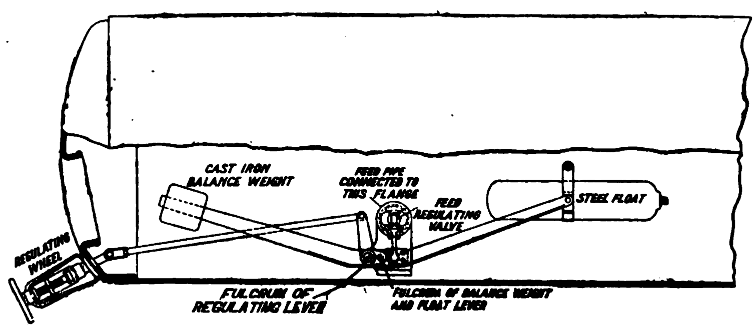


FIG. 100.

Belleville boilers is described in Chapter VIII. Similar apparatus is now generally fitted to all water-tube boilers, and its use has been extended in several cases to water-tank boilers. Two forms of such apparatus are illustrated. Figs. 99 and 100 show the apparatus fitted by Messrs. Thornycroft, while Fig. 101 shows that by Messrs. Laird.

They are very similar in principle, consisting of a hollow float, rising or falling with the water level and actuating a balanced feed-valve, so that when the proper water level is reached, the valve is closed, and

water ceases to enter and vice versa. A detail of Thornycroft's feed-valve is shown in Fig. 99. It is a double beat valve, while that of Messrs. Laird is a circular piston valve. In Fig. 100 the external wheel raises or lowers the fulcrum on which the float lever turns, thus enabling the water level to be adjusted. The external handle in Fig. 101 enables the valve to be moved to and fro in case it sticks.

Surface and bottom blow-out valves.—Blow-out valves are fitted to enable the grease, scum, and other impurities on the water surface, or any dirty water, to be discharged overboard. The usual practice for many years was to fit two screw-down valves as blow-out valves, one, the *surface blow-out or scum valve*, dealing with the water near the water line, and the other, the *bottom blow-out*, dealing with the water at the bottom of the boiler. From the surface blow-out valve, which is always fitted at a convenient height for access, a pipe is carried inside to the central part of the boiler, where it terminates in an open pan placed a little below the water level. When the valve is opened the grease and impurities are discharged through the surface blow-out valve into a pipe led to a sea valve at the bottom of the vessel and thence overboard. The bottom blow-out valve was, in the days of low-pressure steam, the means of filling the boilers with sea-water, and of

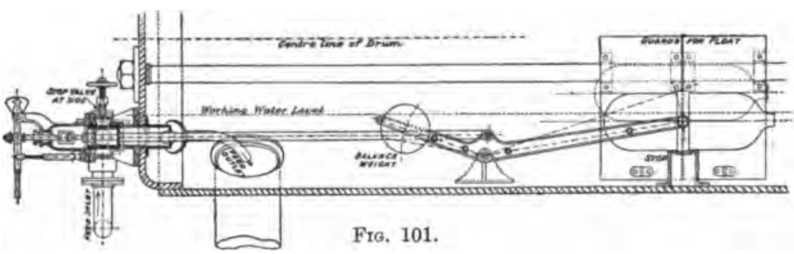


FIG. 101.

brining the boilers when they were fed with salt water, or, in later examples, when salt water was used to make up losses of feed-water. It is an ordinary screw-down valve fitted near the bottom of the boiler, with short internal pipe leading to the lowest part. At this part the densest water and the deposits from the sea-water accumulated, and were blown overboard through the bottom blow out valve. The pipes from surface and bottom blow-out valves led into a common pipe carried to a sea valve, one pipe to each boiler led independently to a guard cock attached to the sea valve. To avoid accident from leaving the blow-out open, a guard is fitted on this cock (Fig. 102), to prevent the spanner from being removed without first closing it; so that when the spanner is off, it is certain that the cock must be shut.

With the more extended use of fresh water for boilers the bottom blow-out cock is of less importance, and as with the higher pressures of steam they are liable to leak and waste fresh water they have not been fitted to naval cylindrical boilers for some years past. Water-tube boilers containing a much smaller quantity of water, which therefore rises in density rapidly when sea water obtains access to it, still have bottom blow-out arrangements fitted to the sediment collector or other part of each boiler. Double valves are generally fitted as a safeguard against leakage. In modern boilers in the Royal Navy,

in lieu of the old blow-out valve, a valve is fitted with a nozzle and hose connection, so that when there is no pressure in the boiler a hose can be connected and water run into the reserve fresh-water tanks, as, for example, when the boilers have been kept full of fresh water, and it is desired to reduce it to working height to enable steam to be raised.

Asbestos packed cocks.—For various purposes cocks are more convenient than valves, the boiler gauge cocks for example, and when such cocks are subject to high pressures, and have at the same time to be not too tight to be readily workable, they generally give trouble by leakage. A device for overcoming this, known as an asbestos-packed cock, is shown in Fig. 102, in which it will be seen that not only are the top

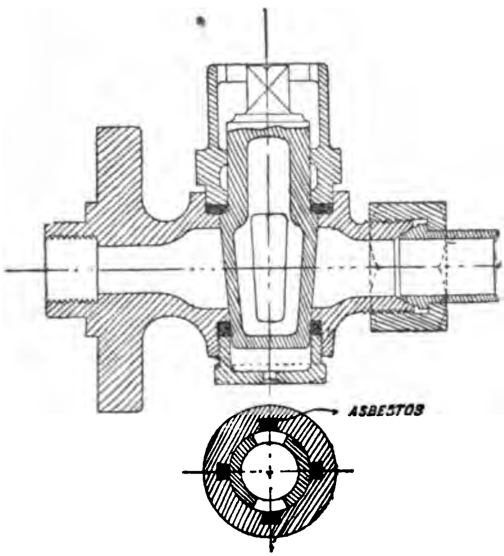


FIG. 102.

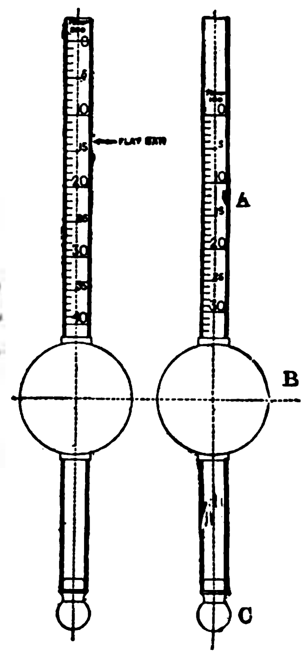


FIG. 103.

and bottom glands packed with asbestos, but longitudinal grooves are formed in the shell, as shown, also packed with asbestos, on which the faces of the plug work. The example shown is a blow-out cock for boilers, the guard which prevents the handle being removed except when the valve is shut, being shown in the figure.

Hydrometer. Hydrometer cock.—The density of the water in the boiler, evaporator, or other part is given by the hydrometer, an instrument of the form shown in Fig. 103, made either of glass or metal. It has a slender stem A, and two bulbs; the larger one, B, containing only air, gives buoyancy, and the smaller one, C, loaded, keeps the instrument vertical in a liquid.

When any body floats freely, the weight of the liquid displaced is

equal to the weight of the body, so that the higher the density of the liquid the less depth will the body sink in it.

Sea water contains $\frac{1}{32}$ part of solid matter, so that hydrometers are often graduated to show densities of 0, $\frac{1}{32}$, $\frac{2}{32}$, and so on—a density of $\frac{2}{32}$ representing the presence of solid matter equal to twice that contained in sea-water. The usual naval hydrometer is graduated in degrees, each degree representing the presence of one-tenth the solid matter in sea-water. Ten degrees, therefore, represent the density of sea-water, or $\frac{1}{32}$ part of solid matter; zero will represent fresh water; and 40 degrees represents a density caused by the presence of four times the solid matter in sea-water.

As the density of water depends on its temperature, a thermometer is really required as well as a hydrometer in order to determine the density with great exactness; but in practice, boiler hydrometers are graduated to suit a temperature of 200° Fahr., which is about the temperature of the water a few seconds after being drawn off for testing. This plan has been found to be sufficiently accurate for the purpose, and is now generally adopted, as it avoids the complication involved in the use of two instruments. The water is drawn off from the *hydrometer cock* (L, Fig. 83), fitted for this purpose, into a long pot called the hydrometer pot, into which the hydrometer is inserted.

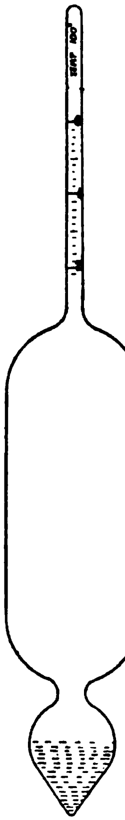


FIG. 103A.—Sensitive Hydrometer.

The brass naval hydrometer has an additional graduation suitable for a temperature of 100° Fahr., placed on the reverse side of the stem, which can be used for taking the density of the feed-water, &c. Care should be taken that the temperature in this case does not differ much from 100°. The two graduations are shown in Fig. 103, and by comparing them it will be seen how important the effect of temperature is, the two scales differing by about 10°.

Sensitive Hydrometer.—With the introduction of water-tube boilers, and especially in the case of the small-tube type, the necessity of using absolutely fresh water became more pronounced, and a more delicate test for density than furnished by the ordinary service hydrometer became necessary. For this reason, the sensitive hydrometer was introduced. This instrument, constructed of glass, on the same principle as the ordinary hydrometer, is graduated as shown in the sketch (Fig. 103A). Each division represents $\frac{1}{100}$th part of the solid matter contained in sea water at 100° F., and the instrument therefore registers $\frac{1}{10}$th of a degree, with a maximum of 2 degrees. Having regard to the influence of the temperature on the density as stated above, the exact temperature for which the hydrometer is graduated must be ensured, and for this purpose a sensitive thermometer is used in conjunction with it.

Nitrate of Silver Test.—If nitrate of silver is added to a solution of chloride of sodium, a white precipitate is formed, the density of

which depends upon the amount of chloride of sodium present; the solid matter in sea water consists largely of chloride of sodium, and advantage is taken of these facts to detect the presence of sea-water, a very minute quantity of sea-water present in the feed-water being indicated by the formation of a white cloud. It should be pointed out, however, that a solution of lime will give what appears to be a similar result, but the colour of the precipitate in daylight is a brownish white, and the difference can usually be detected by experience. In any case, the presence of salt can be confirmed or otherwise by the addition of a few drops of dilute nitric acid, which will dissolve any precipitate formed by a lime solution.

The action of silver nitrate on sodium chloride is also used to give a quantitative measure of the amount of solids in the boiler water. The apparatus consists of a long graduated bottle, and supplies of nitrate of silver solution of definite strength, and of chromate indicator, a 10 per cent. solution of pure neutral potassium chromate. The apparatus determines the amount of chlorine which the boiler water contains per gallon, and the corresponding solid matter is given by a table supplied with the apparatus. The graduated bottle is filled to the zero mark with the water to be tested; add one drop of chromate indicator, and shake the bottle; slowly add the nitrate of silver solution, and continue shaking the bottle. On nearing the full amount of silver solution required the water will turn red for a moment, and then back to yellow when shaken. The moment it turns red and *remains red*, sufficient silver solution has been added. The reading on the graduated bottle at the level of the liquid will then show the amount of chlorine in grains per gallon; and the solid matter can be read off from the table supplied.

Ordinary boiler stop or communication-valves.—These valves are for the purpose of regulating the passage of steam from the boilers to the engines, and to enable any boilers not in use to be shut off from the steam pipes. One of these valves is fitted to each boiler, and connected to the main steam pipe. Its general form and construction are shown in Fig. 104. It is fixed so that the pressure of the steam in the boiler may be inside the valve, which is worked by means of a screw, the spindle passing to the outside of the valve-box through a steam-tight stuffing box and gland as shown. A stop is fitted to prevent the valve turning round when being screwed up.

Self-closing stop-valves.—In warships, in order to lessen the damage resulting from accident to a boiler, as, for example, its being pierced by a shot, the boiler stop-valves are made self-closing. This form of valve is shown in Fig. 105. The valve is simply a non-return valve, the action of the screw outside being only to either keep the valve closed on its seat, or to regulate the amount of opening. In the event of the pressure in any boiler falling from rupture or any other cause, the pressure in the steam pipes would close the valve and isolate the boiler, thus not only minimising the damage, but also the loss of boiler power.

With the ordinary stop-valve, if a hole were made in any boiler, the whole of the boilers in connection would be rendered useless, until the stop-valve of the injured boiler could be closed by hand, which, in all probability, would not be until the steam from all the boilers had

discharged itself through the damaged one, increasing the extent of the disaster, and rendering the ship for the time helpless. Self-acting valves might therefore be of great value in a warship in action.

Valves of this description should always be placed in a horizontal

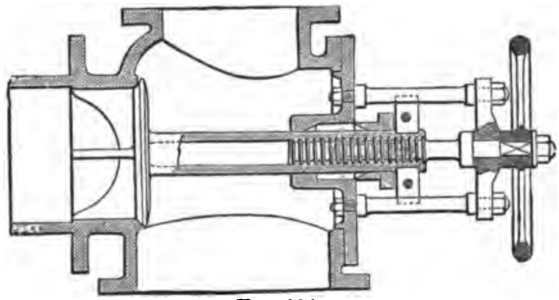


FIG. 104.

position, as when placed vertically, the pulsations of the steam cause them to work up and down on their seatings with violence, and in some cases the valves have been broken from this action.

The continuation of the valve spindle is provided with a cross handle so that the valve can be turned on its seat, and this handle also

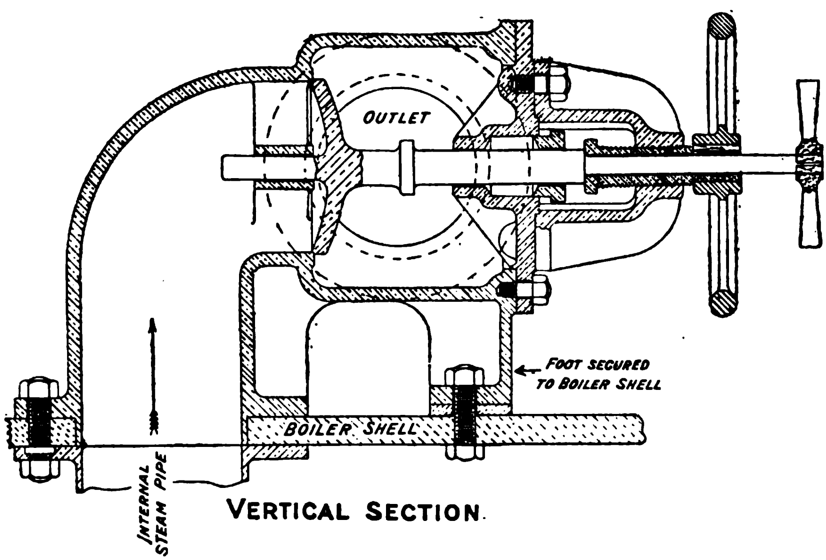


FIG. 105.

enables the valve to be pulled open after the screw has been worked back when opening the valve. This should always be done, as otherwise the friction of the glands will keep the valve closed till the excess pressure inside is sufficient to overcome this friction, when the valve would often open with violence.

Internal steam pipes.—The stop-valves have attached to them thin brass pipes, which are carried inside across the steam chest close to the top of the boiler, and are called the internal steam pipes. They are closed at the end, but have narrow slits cut at the top, which allow the steam to enter the pipe and pass through the stop-valve to the engines. The object of this fitting is to prevent priming as far as possible, or the passage of water through the stop-valves with the steam. By spreading the area of collection of steam, the evaporation is rendered more uniform, and the tendency to priming from the rush of the steam to a single orifice is obviated. A drain hole is fitted in the pipe to prevent accumulation of water.

From the stop-valves on the different boilers steam pipes are carried, which unite at the end of the stokehold in one main steam pipe, through which the steam passes to the engines.

Bulkhead self-closing stop-valves.—In ships of the Royal Navy that have more than one boiler room, the steam pipe from each boiler room is carried independently to the engine-room bulkhead, and at the end of each pipe another self-closing valve is fitted, called the 'bulkhead self-closing valve,' so that in the event of any steam pipe being damaged on the boiler side of this valve, only the boilers in connection with that steam pipe would be put out of action, the others remaining efficient. These bulkhead valves perform for each stokehold the same functions that the boiler self-closing valves perform for each boiler, and are constructed similarly to Fig. 105.

Arrangement of main steam pipes.—The general arrangement of the main steam pipes of vessels in the Royal Navy having several boiler rooms and two engine rooms is shown in Figs. 106 to 108. In each boiler room the steam from each boiler leads through the boiler self-closing valves A into a common pipe which is led to the engine-room bulkhead, where the bulkhead self-closing valve B is fitted. An entirely separate pipe from each boiler room is led to the engine-room bulkhead. On the engine-room side of these bulkhead valves, the steam pipes lead into one common athwartship pipe, of reduced size, communicating with each set of engines. On each branch from this athwartship steam pipe to the main engines an ordinary screw-down stop-valve C is fitted in large vessels, which may be opened or closed by hand from the engine room and also at some position entirely outside the engine room, on one of the ship's decks, by the gear shown in Fig. 107.

This valve is intended for use in case of any accident in the engine rooms causing an escape of steam. Under these circumstances if the discharge of steam is so great as to force the engine-room staff to leave the engine room, they can proceed to the deck position and shut off the steam from the engines.

In the more recent naval engines another stop-valve D is fitted in the athwartship steam pipe at the middle line, worked from either side of the bulkhead, so that in case of any injury to the athwartship pipe the steam can be confined to one side of the ship if necessary, and thus enable one set of engines to be worked.

At the end of the branch steam pipes on each side is the regulating valve E, for the main engines, described in Chapter XV., and fitted close to the high-pressure valve casing.

The additional screw-down valve C is fitted to close with the steam

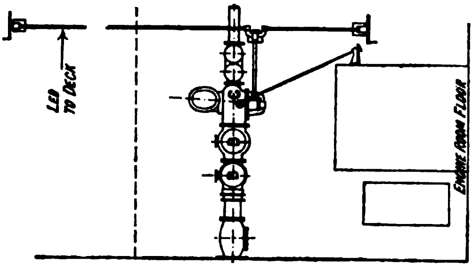


FIG. 107.

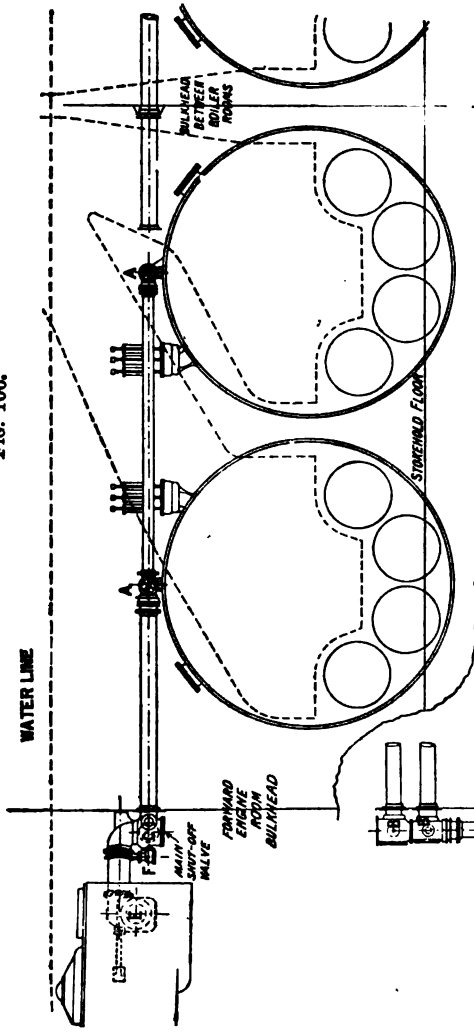


FIG. 106.

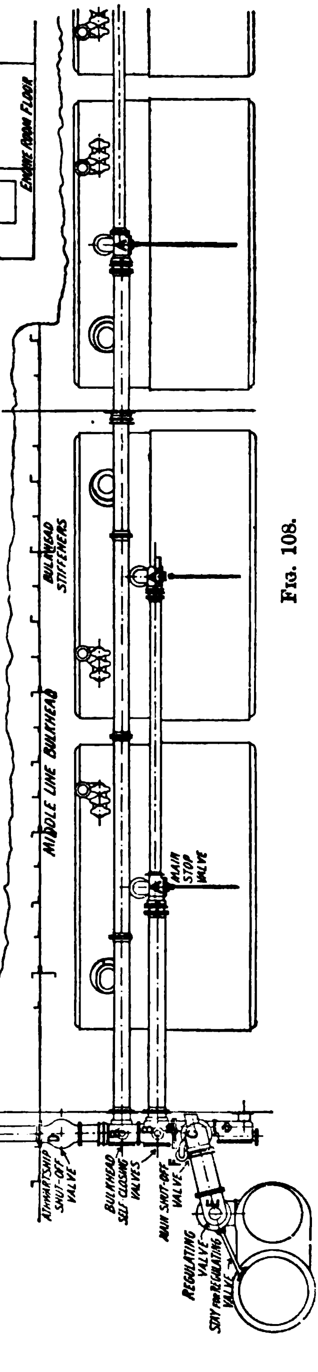


FIG. 108.

pressure to ensure tightness when closed, and to facilitate opening this valve, a small pass-valve F is fitted, by means of which the pressure on each side can be equalised, and moving the valve against the steam pressure is facilitated.

Expansion joints.—As the steam pipes are subject to considerable increase of temperature when they contain steam, they necessarily expand and lengthen, and provision has to be made to enable this expansion to take place without bringing undue strains on any part. In small pipes this can be arranged by bending them sufficiently, so that the pipe at the bend is sufficiently elastic to allow for this elongation and contraction. In large pipes expansion joints have to be fitted which enable one end of the steam pipe to slide in and out of the adjacent pipe as changes of length take place. Fig. 109 shows the usual type of expansion joint formed by a stuffing-box and gland. Care must be taken when such expansion joints are fitted in pipes having a bend, that the unbalanced force is provided for either by struts or safety stays, otherwise the bent part of the pipe may be drawn out of the other. The expansion joints in the arrangement of Fig. 108 are indicated, one being fitted adjacent to each of the boiler stop valves A, and others elsewhere.

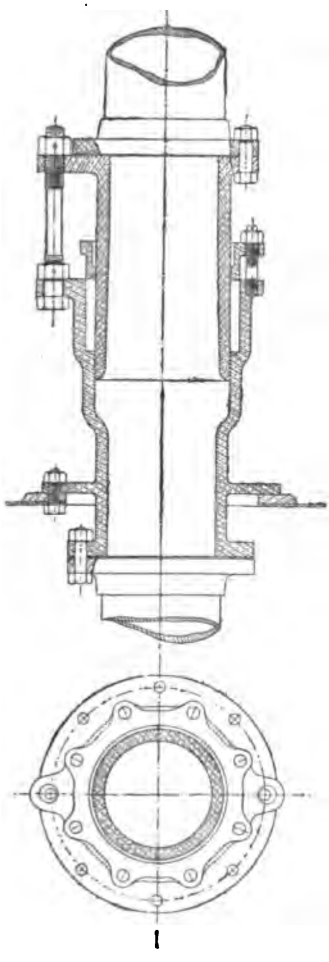


FIG. 109.

Separator.—Before reaching the engines the steam often passes through a separator similar to that shown in Fig. 110. The object is to provide an additional safeguard against priming, by preventing any water in the steam pipes from entering the cylinders. It is divided from the top to nearly the bottom by a diaphragm, D, the steam entering on one side and leaving on the other. Any water that reaches the separator is mostly left at the bottom, only the steam passing on to the cylinders. A drain-cock or valve is fitted, so that the water may be discharged by hand into the hot-well or feed-tank, and returned to the boiler. A non-return valve is fitted at the end of the pipe. The level of water is shown by a glass gauge G.

Automatic separator.—In the fast-running engines of the torpedo-boat destroyer type with water-tube boilers, water is occasionally passed into the steam pipes with the steam which, if it entered the

cylinders, owing to the great speed and light construction of the engines, would probably do considerable damage before being passed through. In the recent vessels of this class, separators with automatic blow-out apparatus are fitted, so that when any quantity of water accumulates in the bottom of the separator by priming or other means, a float is raised, which by a system of levers opens an orifice of considerable area for drainage, so that the water is very quickly got rid of.

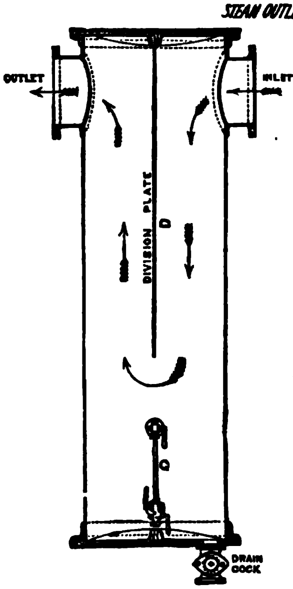


Fig. 110.

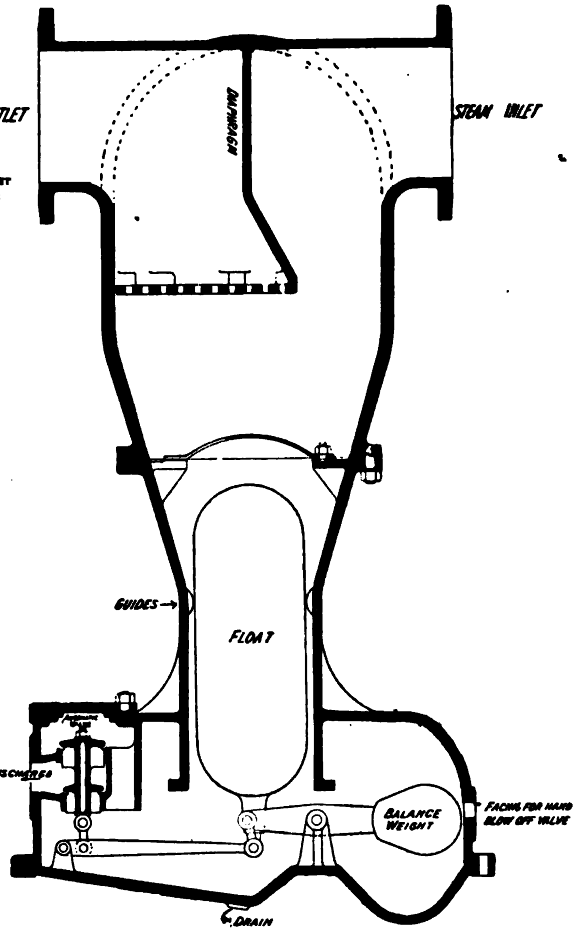


Fig. 111.

With such machinery the old separator with hand-blow-out apparatus is of little or no value. Fig. 111 shows an example fitted in the most recent thirty-knot destroyers. It is a gun-metal casting, provided with the usual diaphragm, while hand-blow-off apparatus is fitted in addition to the automatic arrangement.

Belleville separator.—Figs. 112 to 115 show the arrangement and details of the automatic separator placed in the main steam pipes when Belleville water-tube boilers are fitted. This apparatus, which is found very efficient in practice, consists of a cylindrical separator shown in Fig. 115, with steam inlet at the top and outlet at the side, the division plate between them being of circular form, with closed ends and an opening at the side, as shown by the horizontal section (Fig. 114), so that the steam has to traverse the circular casing before reaching the outlet. A dash plate is fitted below the circular diaphragm, and the water accumulates at the bottom, where an automatic valve worked by steam is fitted, as shown in Fig. 112 at C. To the side of the separator, near the bottom, the steam-tight casting D is bolted, having the double valves and float attached, as shown in Fig. 113.

When there is little water in the separator the float and valves are in the position shown in Fig. 113, the steam valve E being open, and the exhaust valve F shut, so that steam pressure is supplied through the branch B above the valve C, which keeps it tightly shut, as the area acted on by the steam pressure through the branch B is greater than the area of the valve C. When the water rises in the separator it lifts the float G, which actuates the valves E and F, closing E and opening F, which puts the space above the valve in connection with the exhaust, so that the pressure is immediately reduced and the steam pressure below the valve C forces it open, and the water in the separator is blown out. A hand-blow-off cock is fitted also the stop-cock, A, to enable the automatic apparatus to be shut off when required.

As considerable quantities of hot water may be rapidly blown out, the discharge is generally led to the upper part of the tube space of the main condenser, so that the hot water may be cooled before it reaches the feed-tank, otherwise the action of the feed-pumps may be interfered with. To limit the velocity of discharge, the orifice below the valve C, through which the water has to pass, is considerably smaller than the valve.

Auxiliary steam pipes.—Most marine boilers, in addition to the main stop-valves and steam pipes, are fitted with an auxiliary steam service, consisting of small stop-valves, B, Fig. 83, similar in construction to the main valves, leading into a set of steam pipes, which may be used for the auxiliary engines, distilling purposes, &c. These fittings prevent the necessity of filling the whole range of steam pipes with steam when one boiler is in use for distilling or other auxiliary purposes, so that the main valves and pipes are only used for the actual working of the main engines.

The auxiliary steam valves on the boilers have been dispensed with in many ships, especially in those with large numbers of water-tube boilers, branches being led from the main steam pipes to supply the auxiliary steam pipes leading to the auxiliary engines. This arrangement lessens the number of holes required to be cut in the shells of the boilers, and simplifies the arrangement, although when steam is always in one or other of the main steam pipes for long periods in harbour the joints require more care and attention to keep them efficient.

Steam-pipe drains. Steam traps.—The steam and exhaust pipes must be fitted with means of efficiently draining them from water.

Small drain pipes are fitted, which, in important cases, are led to steam traps which automatically drain the steam-pipes, &c., of water.

The action of the trap is as follows (see Fig. 116). The water condensed in the pipes, &c., enters through the inlet orifice A, and fills the

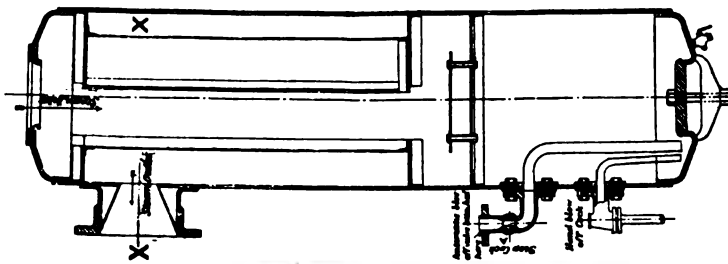


Fig. 115.

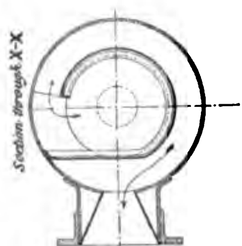


Fig. 114.

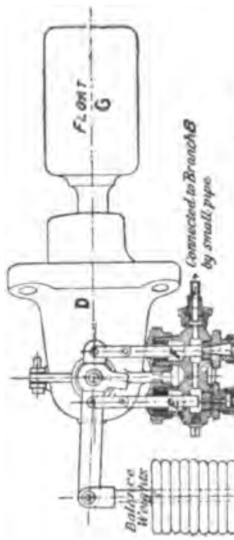


Fig. 113.

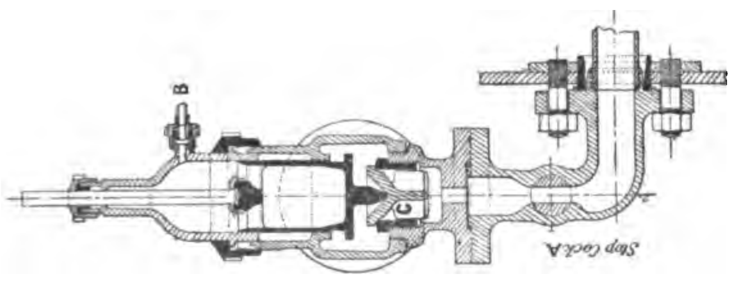


Fig. 112.

annular space G outside the pan until it reaches the level P Q. The water then overflows into the pan and gradually fills it. When the level of the water in the pan reaches a certain height, the pan drops, thus pulling down and opening the valve D, and the water is forced out by the steam

pressure through the orifice F and the outlet passage B to the drain system. When the water level in the pan falls to about X Y, the pressure of the water outside it lifts the pan and keeps the valve closed

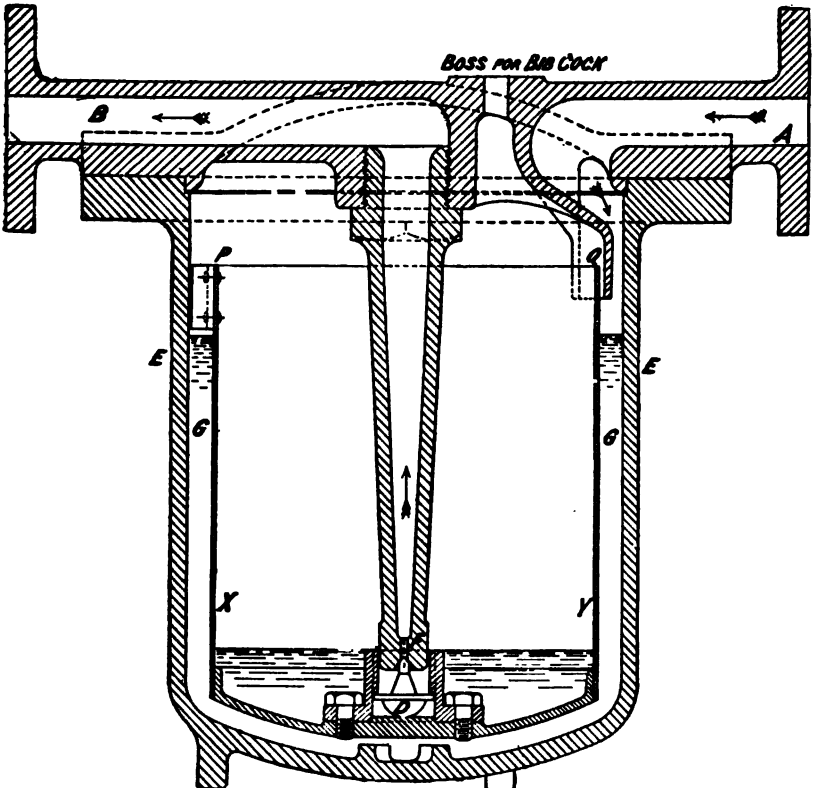


FIG. 116.

until the pan is nearly filled with water, when it is again opened, this action being repeated so long as A is open to steam. A small bib cock is fitted on the cover to ascertain whether the trap is in working order.

Flat glass water gauge mountings.—The usual round thin gauge glasses give trouble with high-pressure steam, owing to frequent fractures, while the water level is often indistinct due to the fairly rapid rate at which these glasses accumulate any dirt present in the water. Klinger's glass, designed to obviate these defects, has therefore been introduced. It consists of a thick flat glass, with smooth front and serrated back, shown in section Fig. 116A, in which figure A and B are the front and back of the mounting, and these are bolted together with the glass and packing, the latter shown by thick lines, between them. The serrations, when clean, cause the water to appear black, as in Fig. 116B. It is found, however, that the serrations soon become dirty, although the glass is comparatively free from failure due to fracture.

Dewrance's flat glass water gauge has also been introduced, and is fitted to modern water-tube boilers for similar reasons. The glasses

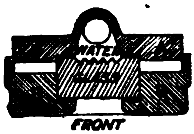


FIG. 116a.

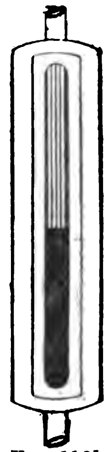


FIG. 116b.

have no serrated surfaces, but are quite smooth, and thus remain clean for longer periods and can be more easily cleaned when dirty.

CHAPTER X.

CORROSION AND PRESERVATION OF BOILERS.

THE question of the durability of boilers is one of the greatest importance as regards the continued efficiency of steamships, and much attention has therefore to be paid to it. The evidence given before an Admiralty Committee on Boilers by leading engineers and chemists showed the great variety of opinions and practice relative to this subject common at that time. This Committee's Reports and further Admiralty experiments cleared up many obscure points, and the methods of treatment subsequently adopted have materially increased the durability of boilers.

The question is one specially important to officers in command; and the leading principles should be clearly understood, enabling the captain and the engineer officer to work together to attain the desired end. Although the care and preservation of the boilers is mostly professional, and in the province of the engineer, many points in their management must be controlled by the captain, and their durability will depend also on his appreciation of the points involved, and knowledge of the subject.

The efficiency of a warship in the present day may be measured largely by that of her machinery, so that if the boilers are injured the evil cannot be estimated by the depreciation in their value alone, as the efficiency of the ship for the purpose for which she is designed is decreased, which is a much more important consideration. Whilst it is important that by proper care and precaution the boilers should be enabled to retain the original working pressures for as lengthened a period as possible, a certain amount of reduction of pressure is always permitted, as they become worn, in order to lengthen their lives, so as to avoid the expense of new boilers and the loss of the vessel's services while new boilers are being fitted.

Within certain limits the initial pressure may be reduced without any very great loss of power for ordinary work, although the consumption of coal will be increased, the increased expenditure of coal being more than compensated for by the ship being kept efficient for probably years longer than would otherwise be the case.

Influence of surface condensation. Early theories as to corrosion.—The introduction of surface condensation at first considerably decreased the durability of boilers, and brought the subject prominently into notice. A special corrosive action was found to take place in boilers supplied with water from surface condensers, and extraordinary cases of rapid decay occurred.

As the tubes in the surface condensers were generally made of copper, it was at first supposed that galvanic action would account for the decay of the boilers, either by the copper condenser tubes and the iron boiler plates forming a great battery, or by the tallow used for lubrication forming fatty acids on decomposition by the heat, and carrying into the boilers particles of copper dissolved from the condenser tubes or feed-pipes. The fact, however, that the condenser tubes and feed-pipes themselves generally appeared to suffer little or no deterioration after considerable periods of work, effectually disposed of this hypothesis.

Although slight traces of copper were found in the specimens of deposit taken from boilers, it was quite insufficient to account for the action produced, and it was also found by experiment that water, even when condensed by tinned or electro-plated tubes, still acted powerfully on the iron, which clearly showed that the action was not due to the contact of the water with the copper condenser tubes.

Influence of vegetable or animal oils. Saponification.—The general opinion of the chemists was that the main causes of the rapid decay of the earlier boilers fed with water from surface condensers were, that the fatty acids evolved by the action known as *saponification* from the heated tallow and vegetable oils at that time used for internal lubrication, were carried into the boilers by the feed-water, and acted directly as corrosive agents on the iron of the boiler plates and stays, and destroyed them. This action was intensified with the superheated steam then used.

Tallow is a compound of fatty acids, chiefly stearic, with glycerine, and when boiled in a solution of soda it is decomposed into stearic acid and glycerine, of which the former unites with the soda, forming a soap, the glycerine remaining free. A similar decomposition takes place when tallow is boiled alone at high pressures, but as no soda is present, the acid remains free; and though these fatty acids are feeble in comparison with mineral acids such as sulphuric, hydrochloric, &c., yet they slowly attack iron and other metals.

The remedy for this is the employment of hydrocarbon or mineral oils for the lubrication of the interior of the cylinders, slides, &c., instead of tallow or oils of vegetable or animal origin, as these mineral oils are not subject to decomposition so as to produce fatty acids. The use of mineral oils for the internal lubrication of engines with surface condensers is now general.

Irregularity of corrosion.—One remarkable feature of this corrosive action was its irregularity. Whilst the boilers of some ships were completely worn out in a very short time, the boilers of other ships employed on the same service, and treated apparently in a similar manner, showed no unusual corrosion. Even in the same boiler some plates have been found to be seriously corroded whilst adjoining ones have been unaffected.

The most serious decay showed itself in the form of *pitting* or local corrosion, deep pits being formed in the plates. This has been attributed to the presence of slag, or to irregularity in the structure of the material, the softer parts being the more readily attacked.

The irregularity of the action was also explained by the fact that clean surfaces would be attacked more readily, and that all parts

covered with a thin film of oxide or scale would be protected. Caustic lime or soda was suggested to be put into the boilers from time to time to neutralise the acids in the water.

Use of wrought-iron.—Wrought-iron was at first universally employed for all parts of boilers, and it was supposed that the irregular corrosion in iron boilers might be due to the manner in which the plates were made, and from which homogeneity cannot be expected. The slag in the puddle-bars is squeezed out more or less according to the amount of work performed on the iron, but it is impossible to be certain that it has been altogether eliminated, and, if not, laminations and blisters in the plate result.

Steel plates.—With steel the case is quite different, as it can be cast in an ingot of sufficient size to form the plate, so that no welding is required, and the ingot has only to be hammered and rolled to form the finished plate, the structure of which should be homogeneous.

For some years, however, steel was looked upon with suspicion and regarded as unreliable for boilers; but the steel now used for this purpose, although containing sufficient carbon to enable it to be fused to insure homogeneity, cannot be hardened, and may be worked with more freedom than iron. The use of mild steel made by the Siemens-Martin process now general, has considerably helped to solve the boiler corrosion question. It is less costly, and stronger than iron, so that boilers may be more cheaply and lightly made. The plates can also be rolled of larger size than with iron, which simplifies construction.

Marine boilers are now made entirely of steel, except the tubes, which are usually of iron in the mercantile marine. In the Navy the tubes are also of steel. The furnaces and internal parts that have to be welded or flanged are made from specially soft steel plates.

Present causes of corrosion in boilers.—The specially severe corrosion which occurred when vegetable or animal oils were used for cylinder lubrication has been obviated by the use of mineral oils for such internal parts.

The principal cause of corrosion in boilers at the present time is the oxidation of the plates, which results from contact with *moisture and air*, either carried in with the feed-water when at work, or existing in the atmosphere when the boilers are empty. This action requires the simultaneous presence of both air and moisture, for neither dry air, nor fresh water thoroughly deprived of air, has any chemical action on steel or iron. Air dissolved in water is especially energetic, and the action is increased by the presence of various chlorides, such as those of magnesium and sodium.

There are other causes of corrosion; for example, hot sea-water, even when entirely deprived of air, has some action on steel and iron. It is stated that at the high temperatures now common in boilers, the chloride of magnesium contained in it is decomposed by the heat and gives off hydrochloric acid, the evolution of acid being accelerated with increase of density. Sea-water should, if possible, never be admitted; but should any obtain access to high-pressure boilers, it is important that sufficient alkali, preferably lime, be admitted to the feed-water to render the water in the boilers slightly alkaline by the litmus test.

A probable minor cause of corrosion in boilers is galvanic action due to differences in the material used in their construction.

Prevention of corrosion when at work.—The admission of air into the boilers is prevented as much as possible, when the boilers are at work, by the separate feed or hot-well tanks described in Chapter XX., with ample surface and other means for the escape of air, and by the fitting of independent feed-pumps, which can be so regulated in speed as to be always fully supplied with water, and never to empty the feed-tank, and so suck in and discharge air into the boilers. The necessity for the complete exclusion of sea-water has already been pointed out; the waste of feed-water should be made good by the evaporators now always fitted together with a reserve of fresh water in tanks. Mineral oils which consist of hydro-carbons only, should be exclusively used for lubrication of all internal parts, air pump rods, and the piston and slide rods which enter the steam spaces of cylinders. Should, however, the boiler water be found by the litmus test to be apparently neutral or to show traces of acidity, it should be made slightly alkaline by the admission of lime or soda into the feed-water.

Zinc protectors.—Besides these items a precautionary means of protection consists in the suspension of slabs of zinc in various parts of the boilers, both below the water-line and in the steam space, in the manner indicated in Fig. 117. If there be any galvanic action the zinc slabs will be attacked instead of the material of the boiler itself. It is important in fixing these zinc slabs that they should be in actual *bright metallic contact* with the material of the boiler and well distributed, so that every portion of the boiler surface is protected. The uniform distribution of zinc slabs over the surface of the boiler is indicated by the positions shown in Figs. 29 and 30, representing the usual arrangement. When properly fitted they produce a beneficial effect, although whether the corrosive action they prevent is entirely galvanic or partly chemical is not fully determined. Zinc being a strongly electro-positive metal, it causes the steel of the boiler to become electro-negative, and induces any corrosive agents to attack itself, leaving the steel uninjured. This action requires the presence of an exciting liquid, so that the zincs have no preservative action when the boilers are empty. The zincs fitted in the steam space act when the boilers are completely filled with water.

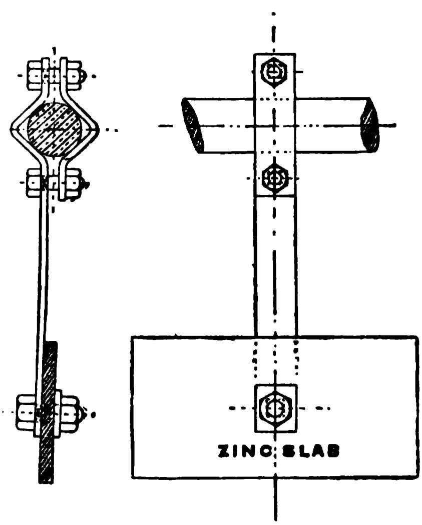


FIG. 117.

Procedure if corrosion is discovered.—When the foregoing precautions are attended to, the decay of water-tank boilers on service is not very great, but if oxidation occurs at any part it will probably be owing to the nearest zinc being too far away from it, requiring re-

arrangement of or additional slabs, or from decay of the zinc, or failure of the metallic connections. As a further precaution the affected part should be carefully cleaned with strong soda solution, and all rust scraped off, and, provided it is not a heating surface, the clean surface coated with a thin coating of Portland cement, which is a substance impervious to moisture.

Effect of intermittent working.—The work of warships is, from its variety and intermittent action, with consequent repeated expansion and contraction, more trying to the boilers than the steady and continuous steaming usual in the ships of the mercantile marine.

Warships on ordinary service only steam at slow speeds, the engines developing, say, from one-fifth to one-twentieth their full power. Only a portion of the boilers is required at one time, and if any are kept empty, unless great care be exercised, moisture will get into them, through the cocks and valves, and this moisture, if atmospheric air be present, is one of the most important causes of corrosion.

General preservation when at work.—Changes of temperature in tank boilers should take place as seldom and gradually as possible. Unless in cases of emergency, steam should be raised slowly, to allow the different parts of the boilers to gradually expand and prevent local straining. Banking fires should be resorted to as rarely as possible, to prevent change of temperature. Any saving of coal that may be gained by frequently banking fires is paid for at the expense of the boilers themselves.

Injury may be done by drawing fires when steaming is over, as large volumes of cold air rush through the boilers, and the sudden contraction that ensues frequently causes leaks, and damages the boilers. If possible, sufficient notice as to the probable length of time that steam will be required should be given, to allow the fires to burn down ; and when the engines are done with, the boilers should be closed up and allowed to cool, so that they contract gradually and prevent undue strains. The furnaces should be cleared out after all has become cool.

Water-tube boilers are less liable to injury from the above causes, and, owing to the rapidity with which steam may be raised in them, the banking of fires is not necessary.

In cases where bottom blow-out arrangements are fitted, the water should never be emptied by the steam pressure, unless on an emergency, but allowed to remain until cool and then run or pumped out. This is a more tedious process, but the efficiency of the boilers is the first point to be considered.

Cleanliness.—As regards the preservation of boilers, their cleanliness is of the first importance. As little oil as possible for internal lubrication should be used, as this, when admitted to the boilers, is deposited on the heating surfaces in the form of a highly non-conducting substance, which often leads to the overheating and absolute failure of the plates and tube ends. Consequent overheating due to such greasy deposits is a fruitful source of wear in boilers.

Much of this oil will float on the surface of the water, so that a slight occasional use of the surface blow-out is desirable. When boilers are intended to be completely emptied, the surface water should always first be blown out, as otherwise the oil floating there, on its

descent through the boiler when emptying, will be deposited on the boiler surfaces.

To reduce the amount of oil entering the boiler, modern ships are now always fitted with filters, through which the feed-water has to pass, and which prevents most of the grease from entering the boilers (see Chapter XXVIII.).

In most tank boilers it is impossible to thoroughly clean the tube plates without drawing the plain tubes. The tubes, being arranged in vertical rows, allow of narrow scrapers being worked vertically between them, and in this way these parts of the boiler can be fairly well cleaned, but it is impossible to thoroughly clean the horizontal spaces between the tubes. As a general rule, each boiler should have many of the tubes withdrawn for cleaning purposes about once in two years, if the conditions of the service on which the ship is employed permit; but the vessel need not be laid up at all for this, as the tubes can be drawn in regular rotation, dealing with a nest at a time.

Care is required in drawing boiler tubes to avoid damaging many of them during the operation, but when properly carried out with steel tubes not more than 5 per cent. need be spoilt in the case of tank boilers. Before replacing the tubes, the ends should be annealed and dressed.

Preservation of boilers when not at work.—Boilers not in use for a lengthened period, in store, or on board ships, may deteriorate very rapidly if proper precautions be not taken. To prevent this if on shore they are carefully dried, and perforated trays containing burning charcoal or coke are placed in them, the boilers being then immediately closed and hermetically sealed, to prevent access of air. The glowing carbon will absorb most of the oxygen of the air in the boiler, and no internal decay will ensue. This is the best method of preservation. Another plan after drying is to insert about $\frac{3}{4}$ cwt. of quicklime per furnace in shallow trays, also a tray of burning coal, well coked, and then close up. The quicklime absorbs any moisture.

If the boilers are on board ship they may be preserved in the same way if there is risk of water freezing, and it is inconvenient to avoid this by means of stoves; but if kept empty on board ships afloat there is always danger of leakage past the sea valves, which would neutralise the preservative measures. Special care will be required to guard against leakage past the sea valves, which must be kept in good order. If steam is required to be raised in any of the other boilers this method becomes impracticable owing to the added danger of leakage past the steam stop and feed valves.

The best method of preservation for boilers afloat is to keep them quite full of fresh water made distinctly alkaline, and by putting on a slight pressure the air escapes through a small cock at the highest part. If possible the water should be heated to expel the air. After using this method the boilers should be emptied and washed out before steam is raised. If steam has to be raised in some boilers and this method is for any reason impracticable, the empty boilers should be kept open and dry and raised above the temperature of the atmosphere by stoves in the ash pits.

Corrosion of water-tube boilers.—In such boilers experience up to the present shows that cleanliness of the surfaces and tubes is the most important element in their preservation. All foreign matter should be

kept out of the boilers, and the system adopted in repairing and cleaning should be arranged so as to avoid the possibility of anything being left in them which might fall into and choke the tubes or obstruct the circulation of water. When a tube fails quickly the cause is generally found to be overheating, due to the circulation being obstructed by the presence of foreign matter in the tube, or to the presence of some saline lime or greasy deposits on the surfaces, obstructing the passage of the heat from the tube to the water. Zinc slabs have been found to exert considerable protective action in water-tube boilers in which fresh and even distilled water has been generally used, and the good contact and renewal of these fittings should therefore be carefully attended to.

Design of water-tube boilers affecting durability.—The result of experience of a large number of ships with different designs of water-tube boilers of the small-tube type in the British Navy is that to prolong the life of the tubes as much as possible: (1) The top of the tubes should not rise much above the water level, i.e. the tubes should be drowned or practically so. (2) Reduced bottom ends of tubes should not be fitted, as the contraction of diameter forms a recess for lodgment of dirt and scale which causes the tube to become overheated and facilitates the choking of the end. It also renders the tube more difficult to search by a cleaning tool. (3) Quick bends of tubes should be avoided, as brushes or clearing tools cannot be readily passed through them, and they are not kept properly clear, their life being thereby shortened. Bends with radius below 12 inches should be avoided if possible.

Many boilers in which the above conditions are not fulfilled are excellent steam producers when new, but their decay is more rapid than in those in which these conditions are fulfilled.

Life of boiler tubes.—Boiler tubes being thinner than the other parts of a boiler require renewing more often. The average life of steel tubes in water-tank boilers is, with careful treatment, about eight years, but is often less.

In water-tube boilers their life varies very greatly. From four to five years is the average life at present in water-tube boilers of the small-tube type, but several cases have occurred where the life has been less than this period. With Belleville boilers at least six years is the probable average life of the tubes, although individual tubes occasionally fail by pitting at an earlier period.

Quality of water used.—Care should be exercised to keep the water used in water-tube boilers as clean and fresh as possible. Severe priming generally occurs if they are worked with water containing much sea water; and as it is very difficult in many water-tube boilers to clean off deposits of lime when once formed, sea water and also shore water containing lime salts should be carefully excluded.

Corrosion is sometimes produced by using certain shore waters. All such waters contain air and other gases dissolved, and these being set free when the water is heated may act on the boiler. Some waters also contain an appreciable amount of matter in suspension, and this will be deposited on the surfaces, and by retaining moisture and keeping the surfaces damp when the water is run off, may cause corrosion. Some shore waters, particularly in volcanic regions, have distinctly acid properties, and these should be carefully avoided.

CHAPTER XI.

EFFICIENCY OF THE STEAM.

THE total amount of energy in the form of heat transferred to the water in the boiler in order to convert it into steam is not given out as mechanical energy at the engine, but only a small portion of it—say, in ordinary cases, from one-twentieth to one-fifth, according to the type of engine. The ratio which the energy exerted by the steam bears to the total amount of energy in the form of heat expended in its generation is called *the efficiency of the steam.*

It was pointed out in Chapter III. that the total heat of evaporation of steam slowly increased as the temperature of evaporation was raised. In other words, the expenditure of heat necessary to produce a given weight of steam from water supplied to the boiler at a given temperature, increases when the pressure and temperature of the steam are increased. The rate of increase in the total heat of evaporation is, however, very slow. For example, the expenditure of heat required to produce a given weight of steam at the pressure of 10 atmospheres is only 1·04 times that necessary to produce an equal weight of steam at the atmospheric pressure, the temperature of the feed-water in each case being 100° Fahr.

Since the difference between the amounts of heat required to produce a given weight of steam at different pressures is so small, the problem of obtaining the greatest possible quantity of work from a given expenditure of *heat*, is reduced practically to the simpler one of obtaining the greatest amount of work from a given weight of *steam*, the difference in the total heat of evaporation at various pressures being so slight that it may be neglected in approximate calculations.

The indicator diagram.—Before dealing further with the question of the efficiency of steam, it will be necessary to explain the diagram known as the indicator diagram, by which the action of steam in the cylinder is best represented. The term 'indicator diagram' is derived from the instrument used in obtaining it in practice.[1]

This diagram is the geometrical representation of the pressure of steam in the cylinder at various points in the stroke of the piston. The diagram, Fig. 118, is a *theoretical indicator diagram* in which the horizontal ordinates represent volumes, and the vertical ordinates pressures. Its area may be calculated from geometrical principles, so that the diagram may be used in theoretical investigations on the power and efficiency of the engines. For a cylinder of given diameter, the volume may be represented by the length of the stroke of the piston.

In Fig. 118, let O P represent the stroke of the piston of the engine,

[1] See Chapter XXVI.

and O A the *absolute* pressure of the steam during its admission. The initial pressure of the steam in the cylinder is never quite so great as that in the boiler, because a portion of the energy of the steam has to be exerted in overcoming the resistance of the stop-valves, steam pipes, ports, passages, &c. In ordinary cases of marine engines working at full power the reduction of pressure due to this cause may be taken to be about one-tenth of the absolute pressure of the steam in the boilers, but under certain circumstances, which will be explained later,[1] the reduction may considerably exceed even this amount.

During admission the steam passes into the cylinder at this reduced initial pressure, and this part of the action of the steam is shown by the line A B, which is commonly known as the '*steam line*' of the diagram. At B, when the piston has traversed the part O N of its stroke, the whole stroke being represented by O P, the admission of steam to the cylinder is cut off by the closing of the steam ports by the valve.

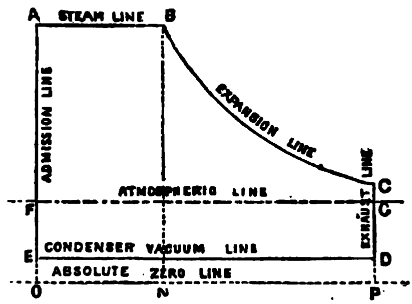


FIG. 118.

The expansion of the steam in the cylinder now commences, and the piston is pressed forward by the expansive force of the steam, the pressure continually diminishing as the piston moves onward to the end of the stroke and increases the volume occupied by the steam. This part of the action is represented by the '*expansion curve*' B C.

The *ratio of expansion* is the ratio between the final volume of the expanding steam, and its volume at the instant of cut-off. In our example $\frac{OP}{ON}$ is the ratio of expansion.

The laws according to which the pressure of steam diminishes during its expansion vary according to the conditions under which the expansion takes place, and the initial state of the steam.[2] For elementary purposes, however, it is sufficient to assume the simple approximate law that the *absolute* pressure will vary inversely as the volume—i.e. when the volume is doubled, the *absolute* pressure falls to one-half, when it is trebled to one-third, and so on.

When the piston arrives at the end of its stroke and the expansion is finished, the communication with the receiver or condenser is opened, the steam escapes, and the pressure falls to P D, the constant back pressure which acts against the piston during the whole of the return stroke. Fig. 118 is the diagram of a condensing engine, and in this case the line D E is technically called the '*vacuum line*' of the diagram. O P is the '*zero line*,' or line of no pressure, from which all absolute pressures are measured. F G is called the '*atmospheric line*,' O F representing the pressure of the atmosphere.

It is important to remember that in all investigations on the action

[1] See Chapter XXVI. [2] See Chapter XII.

of steam, the *absolute pressures*, or the pressures measured from the zero line must be taken, and not the pressures indicated by ordinary pressure gauges, which simply represent the excess of the steam pressure above that of the atmosphere.

Area of the indicator diagram.—*Mechanical work* is produced by the exertion of a force through a space, and the mean value of the force multiplied by the space through which it acts, gives the amount of mechanical work done. In the case of a steam-engine the space is represented by the distance through which the piston travels in a given time, and the force is the excess of the average forward pressure exerted by the steam on one side of the piston, during its admission and expansion, above the average back pressure it exerts on the other side of the piston whilst being discharged from the cylinder.

It is easy to see that the area of the diagram A B C D E represents on some proper scale the work done by the steam on one side of the piston during the double stroke described above.

The greater, therefore, the area of this diagram obtained from a given weight of steam, the greater will be the efficiency of the steam.

The area may evidently be increased in two ways, namely (1) by reducing the back pressure—i.e. by lowering the line E D—or (2) by increasing the mean height of the upper part of the diagram obtained from the same weight of steam used; (1) is effected by means of the condensation of steam and (2) by means of its expansion.

Increase of efficiency due to condensation.—We will now consider the increase of efficiency of the steam due to the application of the principle of condensation.

In the non-condensing engine (often called by the misleading term 'high-pressure engine') the steam, after having done its work in the cylinder, escapes to the atmosphere, and the back pressure is not less than from 3 to 4 lbs. per square inch above the atmosphere, corresponding to an absolute pressure of from 18 to 19 lbs. per square inch, or a temperature of about 224° Fahr., and with contracted exhaust passages and quick-moving engines it is sometimes much higher. When a condenser is used so that a partial vacuum is formed in the cylinder behind the piston, the back pressure is only 3 to 4 lbs. *absolute*, corresponding to a temperature of 140° Fahr. to 150° Fahr. In this case, where the back pressure is less than that of the atmosphere, the difference between the back and atmospheric pressures is technically called the '*vacuum in the cylinder.*' The amount of vacuum in a cylinder or condenser is generally measured in inches of mercury.

Imagine two engines, one condensing, the other non-condensing, working with steam of 60 lbs. per square inch above the atmosphere, or 75 lbs. absolute, the initial pressure and ratio of expansion being the same in each case. Then the steam line, until the end of the forward stroke, would be the same both in the condensing and non-condensing engines, the only difference in the two cases being in the position of the line of back pressure.

The indicator diagrams, Fig. 119, represent the action of the steam in the two engines under consideration, the corners of the diagrams being here rounded off as is the case in actual diagrams, due to the gradual opening and closing of the ports by the slide valve, instead of the sudden opening and closing assumed in theoretical diagrams.

In the non-condensing engine the back pressure line D E, Fig. 119, will be about 3 or 4 lbs. above the atmospheric line H K. In the condensing engine the pressure at the end of the stroke falls considerably below the atmospheric pressure, and the back pressure will only be 3 to 4 lbs. absolute, or, say, 12 to 11 lbs. below the atmospheric line, as shown by the line F G, so that by the application of the condenser the work done by the same weight of steam is increased by an amount represented by the area E D F G.

The forward pressure would be the same in each case, but in the non-condensing engine this would be resisted by a back pressure of 18 to 19 lbs. per square inch; whereas with the condenser, the pressure resisting the forward motion would be only 3 to 4 lbs. absolute. It is clear that, in a non-condensing engine, the cut-off should never be early enough to cause the pressure of steam to fall below that of the atmosphere before the completion of the stroke; for this would necessitate the latter part of the stroke being performed by the expenditure of work accumulated in the moving parts during the earlier part of the stroke, and would probably cause difficulty in starting, and prevent smoothness and regularity in working.

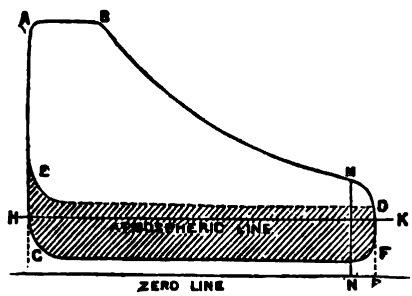


FIG. 119.

It is evident that an equal quantity of steam would be used by both engines, because the absolute pressure of the steam N M just before release, which represents the steam used, is the same both in the condensing and the non-condensing diagrams.

Application of formula for maximum efficiency.—A very important theorem and formula of the science of thermo-dynamics will enable this increase of efficiency to be shown. It is as follows, viz.:—

The greatest possible efficiency of any heat engine depends on the difference between the initial and final temperatures at which it is worked, and is represented by the formula—

$$\text{Maximum efficiency} = \frac{T_1 - T_2}{T_1} = \frac{t_1 - t_2}{t_1 + 461}$$

where T_1 = initial absolute temperature $= t_1 + 461$
T_2 = final absolute temperature $= t_2 + 461$
t_1 = initial temperature in degrees Fahr.
and t_2 = final temperature in degrees Fahr.

We will apply this formula to the cases of the two engines above referred to. The temperature corresponding to 75 lbs. pressure absolute is 307° Fahr., while the corresponding temperature for 18 to 19 lbs. pressure is about 224°, and for 3 to 4 lbs. 150° Fahr.

The non-condensing engine would therefore be working between the limits 307° and 224°, whilst the limits in the case of the condensing engine would be 307° and 150°. The relative maximum efficiency of

the condensing to that of the non-condensing engine would therefore be—

$$\frac{307-150}{307-224}=\frac{157}{83}$$

or nearly two to one, so that the efficiency of the steam could be nearly doubled by the addition of a suitable condenser. It should be noticed from the form of the equation that the relative gain is much greater at low pressures than at high pressures.

The following table given by Professor Cotterill shows the calculated consumptions of steam per hour if the engine were perfect, both in the case of a condensing and of a non-condensing engine, and clearly shows the gain in efficiency due to the condensation, and also that the percentage of gain is greater at low than at high pressures.

Pounds of Steam per I.H.P. per hour.

Initial pressure in atmospheres	Condensing	Non-condensing	t_1 in degrees Fahr.
	lbs.	lbs.	
2	11·2	50·4	249°
4	9·2	24·8	291°
6	8·3	18·9	318°
8	7·7	15·9	340°

In the condensing engine it was assumed that the feed-water was taken from the condenser at a temperature of 100° Fahr., and in the non-condensing engine that the exhaust steam had been used to raise the temperature of the feed-water to about 212° Fahr.

Steam efficiencies of actual engines.—Average values for the steam efficiencies, as calculated from trial records of a number of naval vessels, are given in the table below :—

Type of vessel	Steam Pressure at engines, lbs. per sq. in.	Steam efficiency per cent. at about		
		Half power	¾ power	Full power
Battleships . .	250	14·6	14	13
" . .	200	14·9	15 2	14·1
Large cruisers .	250	14·25	14·2	14
" . .	200	14·5	14·9	13·8
Small cruisers . .	250	14·2	14	13

The efficiency of the steam is deduced from the following data recorded on the contractor's trials of new ships :—

(1) Quantity of steam used per hour in pounds.
(2) Average steam pressure in pounds per square inch.
(3) Average vacuum in inches of mercury.
(4) Average I.H.P. developed.

(1) is measured by collecting, in special measuring tanks, the steam which has passed through the main engines after being discharged as water from the condensers. The steam used by the main engines is, where possible, kept separate from that used by auxiliary machinery,

the latter being measured separately. (2), (3) and (4) are recorded at regular intervals, and the average of the records used in the subsequent calculations.

Actual numerical example.—The following example will show the method of calculating the steam efficiency from the trial records, the data having been recorded on the machinery trial of a war vessel :—

Quantity of steam passing through the main engines in ten minutes, as measured by the weight of water collected in measuring tanks = 32,500 pounds.

Average I.H.P. = 13100.

Average steam pressure = 245 pounds per square inch.

Average vacuum in condensers = 25 inches.

From these records we can deduce the efficiency of the steam thus :—

The total heat of 1 pound of dry saturated steam at a pressure of 245 pounds per square inch from feed water at 32° F.

$$= 1205{\cdot}3 \text{ British thermal units.}$$

Temperature corresponding to a vacuum of 25 inches = 133·2° F.

Hence, heat available per pound of steam, between the working limits of pressure and temperature

$$= (1205{\cdot}3 + 32 - 133{\cdot}2) = 1104{\cdot}1 \text{ British thermal units.}$$

Heat available in 3250 pounds of steam

$$= 1104{\cdot}1 \times 3250 \text{ British thermal units.}$$

And this quantity of steam is capable of performing, as indicated work,

$$13100 \times 33000 \text{ foot pounds} = \frac{13100 \times 33000}{778} \text{ British thermal units.}$$

Therefore the steam efficiency

$$= \frac{13100 \times 33000}{778 \times 3250 \times 1104{\cdot}1} = 15\tfrac{1}{2} \text{ per cent., nearly.}$$

Sources of loss.—The principal sources of loss of steam efficiency are :—

(1) The loss due to the large amount of heat contained in the exhaust steam which is wasted in uselessly heating the circulating water in the condensers. This loss is unavoidable, and is common to all types of steam-engine.

(2) Losses due to condensation of steam in the cylinders during admission, accompanied by re-evaporation, and consequent loss, during exhaust.

(3) Losses due to conduction, radiation, wire-drawing, and imperfect expansion of the steam.

CHAPTER XII.

EXPANSION OF STEAM.

We will now consider the increase in the efficiency of the steam obtained by utilising its property of expansion, the second of the two means of increasing efficiency referred to in the last chapter.

Referring again to an indicator diagram, Fig. 120, it will be seen that the pressure represented by the mean height of the line A B C can be divided into two distinct parts; first, the pressure during admission while the steam is passing from the steam pipes into the cylinder, represented by the height of A B, and second, the diminishing pressure during the expansion of the steam when its flow into the cylinder has been cut off, this diminishing pressure being represented by the expansion curve B C. It is clear therefore that, as there is no further expenditure of steam during this part of the stroke, the amount of work done during expansion is so much gain, and the amount of work obtained from a given weight of steam, and hence the efficiency of the steam, is increased.

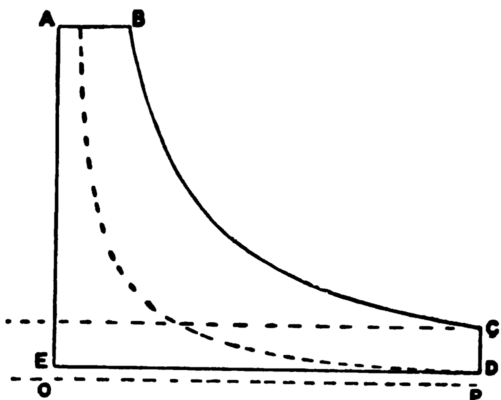


Fig. 120.

Limit of useful expansion.—It is clear from the diagram that energy would continue to be exerted by the steam whilst its forward pressure was greater than the back pressure P D, so that in order to get the greatest possible quantity of energy exerted by the steam, the expansion should be such that the forward pressure becomes so far reduced as to be just equal to the back pressure; i.e. the cut-off should be so early, that the expansion curve will, at the end of the stroke, just fall to the back pressure line D E, as indicated by the dotted curve in the diagram, so that there is no sudden fall of pressure on exhaust.

In a steam-engine a portion of the work obtained from the steam is expended in overcoming the friction of the working parts of the machinery, so that even theoretically, in order to obtain the greatest possible amount of *useful work* from a given quantity of steam, the

expansion should only be carried out until the pressure in the cylinder is so reduced as to be just equal to the back pressure, *plus* a pressure equivalent to the friction of the mechanism.

In practice it is not possible to carry out the expansion, efficiently, to so great an extent as this, and it must only be taken as a theoretical statement of what might be the case if the steam were expanded in a perfectly non-conducting cylinder, and as the condition to which we must endeavour to approximate as closely as possible, by suitable appliances to existing engines.

Illustration of gain by expansion.—By inspection of an indicator diagram it will be obvious that this increase of efficiency due to expanding steam becomes greater as the initial pressure and amount of expansion are increased. This is illustrated in a simple manner by means of a theoretical indicator diagram, in which the pressure is assumed to vary inversely as the volume occupied by the steam, and a cylinder in which different pressures of steam can be admitted and any desired cut-off obtained, the back pressure being constant.

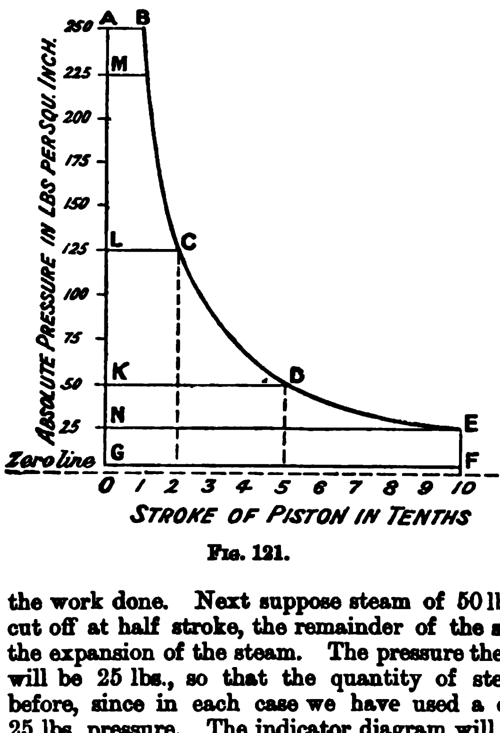


FIG. 121.

Suppose steam of 25 lbs. absolute pressure per square inch be admitted to this cylinder throughout the entire stroke, represented by G F, Fig. 121, and let O G be the constant back pressure, then the indicator diagram will be N E F G, and the rectangle G E will represent the work done. Next suppose steam of 50 lbs. absolute admitted and cut off at half stroke, the remainder of the stroke being completed by the expansion of the steam. The pressure then at the end of the stroke will be 25 lbs., so that the quantity of steam used is the same as before, since in each case we have used a cylinder full of steam of 25 lbs. pressure. The indicator diagram will be K D E F G, so that, subtracting the common area G E, the increase of area representing additional work done is K D E N, and similarly for higher pressures.

The theoretical diagrams for two higher pressures—viz. 125 lbs. initial pressure with cut-off at $\frac{2}{10}$ stroke or ratio of expansion of 5, and 250 lbs. with cut-off at $\frac{1}{10}$ stroke, or ratio of expansion of 10—have also been drawn on the same figure for comparison. It will be seen that as the initial pressure and ratio of expansion are increased, the greater is the gain in the area of the diagram, so that it is clear that theoretically the greater the pressure and number of expansions, the greater is

the efficiency—i.e. the greater the amount of work obtained from a given weight of steam.

It should be noticed, however, that the higher the initial pressure becomes, the smaller is the gain by a given increase of pressure; for example, compare the gain by increasing from 25 lbs. to 50 lbs. represented by K D B N with the gain by increasing from 225 lbs. to 250 lbs. represented by the much smaller area M B.

General conclusions.—By examining the formula for maximum efficiency of the theoretical heat engine,

$$\text{Maximum efficiency} = \frac{t_1 - t_2}{t_1 + 461}$$

the increase in maximum efficiency due to increase of pressure may also be seen, for this expression may be written

$$1 - \frac{t_2 + 461}{t_1 + 461}$$

which becomes greater as t_1 is increased and therefore also as the pressure is increased.

It was shown in the last chapter that the lower the back pressure the greater was the efficiency, so we learn that the higher the mean forward pressure obtained from a given weight of steam, and the lower the back pressure, the greater is the efficiency of the steam.

The initial temperature and pressure of the steam are limited by considerations of the strength and safety of the boilers, cylinders, and other parts exposed to steam pressure, and also the continued efficiency of all working parts exposed to these high temperatures.

In Watt's time, workmanship and knowledge of the strength of materials were not in such an advanced state as at the present, so that most engineers of that time were necessarily very cautious in the adoption of high pressures, and relied more on obtaining a low back pressure. As experience was gained, the pressures at which boilers were worked were gradually increased, and of late years the advances in that direction have been great, and there has been much gain in economy from the high pressures and rates of expansion now in general use.

In the preceding explanation the expansion of the steam is supposed to be in accordance with the hyperbolic law, which is approximately the case in practice, and a table showing the gain per cent. at various rates of expansion can easily be deduced on this assumption. There are other possible curves of expansion of steam, however, depending on its treatment during expansion (see later in this chapter).

Numerical results for gain by expansion.—The following table has been calculated on the assumption that the expansion curve is a certain curve, which will be explained later, called a 'saturation' curve—i.e. it represents the steam as always in a state of saturation whatever the pressure may be. This curve always falls slightly below the hyperbolic curve. On this assumption it shows how the amount of work done by one pound of steam is augmented as the initial pressure and ratio of expansion are increased. Since the total heat of steam is practically the same at all temperatures, the increase in the performance of work may be taken to represent very nearly the theoretical increase in efficiency due to the increased expansion.

The pressure in each case at the end of the expansion is supposed to be the same, viz. 10 lbs. per square inch absolute, and the back pressure 3 lbs. per square inch absolute.

Initial absolute pressure	Relative volume[1]	Specific volume[1] in cub. ft.	Ratio of expansion	Mean absolute pressure in lbs. per sq. in.	Mean effective pressure in lbs. per sq. in.	Relative indicated horse-power
10	2368	37·8	1·0	10·0	7·0	100
20	1231	19·7	1·9	17·2	14·2	289
40	643	10·3	3·7	24·5	21·5	538
60	434	7·0	5·4	29·0	26·0	716
80	338	5·4	7·0	33·3	30·3	901
100	270	4·4	8·6	36·2	33·2	1033
150	184	2·96	12·7	39·6	36·6	1195
200	142	2·27	16·6	42·5	39·5	1340
250	114	1·83	20·6	44·8	41·8	1459
300	96	1·54	24·5	46·4	43·4	1544

[1] See definitions later in this chapter.

It will be seen that by increasing the initial pressure of the steam from 20 to 80 lbs. per square inch absolute, the work done per pound of steam is increased more than three times, whilst at 100 lbs. pressure it is three and a half times, and at 200 lbs. nearly five times as great as at 20 lbs. pressure absolute—or, say, 5 lbs. above the atmospheric pressure, which was the ordinary working pressure in the early days of steam navigation—the heat required to produce the steam being but little different in each case.

Comparison of throttling and expansive working with simple engines.—The economy due to the expansive working of steam in simple engines used to be practically illustrated by taking indicator diagrams when working expansively and throttled respectively. First, a diagram was taken working expansively, noting the revolutions per minute the engines were making. Secondly, the expansion gear was put out of operation, or the link motion put into full gear, and then the throttle valve partially closed until the revolutions were the same as before. The two diagrams were found to be as shown in Fig. 122. The revolutions being the same, the work done will be the same, and consequently the areas of the two diagrams must be equal. The expansion diagram is represented by the full lines, and the throttled diagram by the dotted lines.

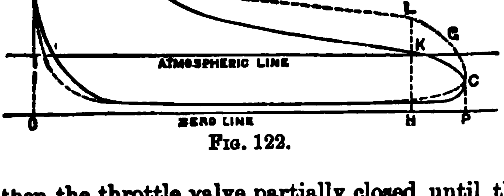


FIG. 122.

The quantity of steam used is represented by its pressure at the

end of the stroke, when the cylinder may be considered to be full of steam at its final pressure, and we therefore see that the amount of steam required to perform a given quantity of work when used expansively is less than when it is throttled and no expansion employed. If we take any point, H, of the stroke just before release commences, it will be seen that the quantities of steam required when working expansively and throttled, will be approximately proportional to the absolute pressures H K and H L, respectively. As less steam is used when working expansively the vacuum will be better, unless the quantity of condensing water is increased when working throttled, which would augment the work done by the pumps, and thus further decrease the efficiency of the engine.

Experiments showing the gain in economy by using high-pressure steam and expansion in a simple engine.—Among many such experiments and tests, a careful series made on the engines of the United States ships 'Bache' and 'Dexter,' with cylinders of from 25 inches to 26 inches diameter, and 2 feet to 3 feet stroke, showed the gain in economy that follows the use of high-pressure steam worked expansively, as compared with steam at a lower pressure worked at a reduced rate of expansion.

In the case of the 'Bache,' the most efficient rate of expansion with the steam of about 80 lbs. pressure when the cylinders were jacketed was about five times. Above this expansion the consumption of feed-water per I.H.P. per hour increased. The trials showed that within the limits that would occur in practice, in case of reduced power being required, the initial pressure should not be reduced and the steam worked at a less rate of expansion, but that the original pressure should be maintained with the higher grade of expansion.

In two expansion trials, with about 81 lbs. pressure, the expenditure of feed-water was 24 lbs. per I.H.P. per hour with $8\frac{1}{2}$ expansions, and 27 lbs. with $12\frac{1}{2}$ expansions, whilst when the initial pressure was reduced to 30 lbs. per square inch, and the expansions to $2\frac{1}{2}$, the consumption of feed-water rose to 34 lbs. per I.H.P. per hour (see columns 0, 2, and 3 of table in Chapter XIII.).

The results from the 'Dexter,' with cylinders not jacketed, were very similar.

Reduced power working.—These results are important with regard to the machinery of warships, which on ordinary service is generally worked at reduced power. It is clear from the figures given, that with engines of the kind experimented on, *it is desirable, for the sake of economy, to work expansively to the greatest extent practicable.*

These results were obtained in simple engines with moderate steam pressures, and although there is reason to believe that the fall in economy when working throttled, as compared with working at a higher rate of expansion and pressure, is not so great with triple-expansion engines as indicated in these experiments, it is still appreciable; so that within ordinary limits, in all engines, the reductions of power should be obtained by linking-up, keeping the steam pressure as high as possible.

The boiler pressure should therefore not be higher than corresponds to the pressure which can be carried in the cylinders, and throttling

should be avoided as much as possible, as it unnecessarily strains the boilers and steam pipes.

When working at reduced powers, the linking-up or other expansion gear should be set to the highest grade at which it can be worked with the regulating valve wide open, or nearly so, and the pressure of steam then kept as high as possible in the slide casings. This statement, however, requires the qualification that the boiler pressure should never be kept lower than that necessary for handling the engines readily, so as to be prepared for the emergency of stopping and starting the engines.

Practical limit to amount of economical expansion.—It will be noticed from the figures given for the actual trial of the 'Bache' that beyond a certain amount of expansion, the consumption of steam increases, which shows that in the ordinary reciprocating engine there is a practical limit to the attainment of the economy which the theoretical diagram indicates. The reason for this will be explained later.[1]

James Watt, and expansion.—Though it is only in comparatively recent years that much attention has been devoted to the development of high rates of expansion of steam in order to attain economical working, we find that in 1769, James Watt indicated the gain that would ensue from the utilisation of the expansive power of steam, and published a body of principles expressing the conditions necessary for the efficient and economical working of the steam-engine. It is remarkable to note how sound these conclusions were.

It will be interesting to state Watt's principles in his own words. He says :—

My method of lessening the consumption of steam and consequently of fuel in fire engines consists of the following principles :—

First.—That vessel in which the powers of steam are to be employed to work the engine, which is called the cylinder in common fire engines, and which I call the steam vessel, must, during the whole time the engine is at work, be kept as hot as the steam that enters it, first by enclosing it in a case of wood or other material that conducts heat slowly; secondly, by surrounding it with steam or other heated bodies; and thirdly, by suffering neither water nor any other substance colder than the steam to enter or touch it during that time.

Secondly.—In engines that are to be worked wholly or partially by the condensation of steam, the steam is to be condensed in vessels distinct from the steam vessels or cylinders, though occasionally communicating with them. These vessels I call condensers, and whilst the engine is working, these condensers ought to be kept as cold as the air in the neighbourhood of the engines, by the application of water or other cold bodies.

Thirdly.—Whatever air or other elastic vapour is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the steam vessels or the condensers by means of pumps wrought by the engines themselves or otherwise.

Fourthly.—I intend in many cases to employ the expansive force of steam to press on the pistons, or whatever may be used instead of them, in the same manner as the pressure of the atmosphere is now employed in ordinary fire engines. In cases where cold water cannot be had in plenty, the engines may be wrought by this force of steam only by discharging the steam into the air after it has done its office.

Lastly.—Instead of using water to render the pistons and other parts of

[1] See under 'Liquefaction in Cylinders' in this chapter.

the engine air and steamtight, I employ oils, wax, resinous bodies, fat of animals, quicksilver, and other metals in their fluid state.

During the same year (1769) Watt invented the cutting off of the admission of steam, so as to make it work expansively, but he did not use it till 1776, and only published it in 1782, when he patented it together with his invention of the double-acting engine.

Before proceeding further it will be necessary to define a few terms that will often recur, as it is important that their meanings should be clearly understood.

Relative volume.—By relative volume is meant the ratio of the volume of the steam produced to that of the water from which it was generated.

Specific volume.—The specific volume of steam is the volume, in cubic feet, of one pound of steam at any given pressure.

Saturated steam.—In all gases the density, pressure, and temperature are connected together by certain fixed laws, so that if any two of them be known, the third can be determined. In the case of steam, or any other vapour, in contact with the liquid from which it is generated, there is, for each temperature, a corresponding density, which is the greatest density the vapour can have without its being partially, or wholly, condensed into the liquid form. Consequently for each temperature there is a maximum pressure which the vapour can exert.

A vapour which is at the maximum density and pressure corresponding to its temperature is called '*saturated vapour.*' It is then just at the point of condensation, and any increase of pressure or decrease of temperature will cause some of the vapour to be condensed. Steam, therefore, at any given pressure is said to be *saturated* when it is at its maximum density consistent with its remaining as vapour. Saturated steam is often called *dry steam*, because it is pure steam without any admixture of liquid water.

Formulæ connecting pressure and temperature.—The relations between the pressure and temperature of saturated vapour are very complicated, and, for all practical purposes, the required results are best taken from tables showing the properties of steam (Chapter III.). Many formulæ have been suggested to represent these relations, but most of them are only of theoretical interest. All the most accurate formulæ are of logarithmic form. The best formula to use in the absence of tables is the following :—

$$\log p = 5\,\frac{t - 212}{t + 367} + \log 14{\cdot}7$$

where p = absolute pressure in pounds per square inch
and t = temperature in degrees Fahrenheit.

This represents the experimental results with fair accuracy.

For theoretical work the following formulæ, which are more exact in form, are most useful.

Suppose t the temperature of the boiling-point on Fahrenheit's scale,

T the absolute temperature of the boiling point, $= t + 461$, and p = absolute pressure of the steam in pounds per square inch,

Then,

$$\log p = \mathrm{A} - \frac{\mathrm{B}}{\mathrm{T}} - \frac{\mathrm{C}}{\mathrm{T}^2} \text{ (Rankine's formula)}$$

$$\log p = a - \frac{b}{\mathrm{T}} - c \log \mathrm{T} \text{ (Dupré's formula),}$$

A, B, C and a, b, c, being constants.

The preceding two formulæ are complex in character, and their calculation would be tedious, but they are of great use in theoretical investigations, the latter being the more accurate.

For practical purposes the following roughly approximate formula may be used :—

$$p = \left(\frac{t + 40}{147}\right)^5$$

t = temperature of boiling-point in degrees Fahrenheit, p = absolute pressure in pounds per square inch.

This is nearly correct for absolute pressures between 6 and 60 pounds per square inch, while for pressures near that of the atmosphere the index becomes 5·5 instead of 5, and for high pressures the index becomes 4·5. This formula is very useful as showing in a form which can be easily appreciated the very small increase of temperature which takes place as the pressure of saturated steam is increased.

Formulæ connecting pressure and volume.—The density of a vapour is measured by the space occupied by a given weight, and the volume of one pound of saturated steam as obtained by direct experiment, may be calculated by the approximate empirical formula given by Fairbairn :—

$$v = \cdot 41 + \frac{389}{p + \cdot 35}$$

where v = volume of one pound in cubic feet, or *specific volume*, and p = absolute pressure in pounds per square inch.

This formula gives results too large when the pressure of 100 lbs. is exceeded.

The formula may be written :—

$$(p + \cdot 35)(v - \cdot 41) = 389$$

which indicates an easy way of forming the curve representing the relation between p and v, viz. by determining one point on it by calculation, and drawing an hyperbola through this point with axes distant ·41 cubic feet to the right, and ·35 lbs. per square inch below the original axes. The hyperbola referred to the original axes will be the required curve (see Fig. 123).

The volume of one pound of saturated steam, at any given absolute pressure, may also be calculated from the formula.[1]

$$p\,v^{\frac{17}{16}} = 475$$

where v = volume in cubic feet

and p = absolute pressure in pounds per square inch.

Superheated steam.—If the steam be removed from contact with the water from which it is generated, and additional heat be applied, the pressure being kept constant, its volume and temperature increase,

[1] A more accurate formula is $p\,v^{1 \cdot 0646} = 479$.

as pointed out in Chapter III., and the steam becomes *superheated*; that is, it contains more heat than that necessary to keep it in a state of saturation at the given pressure. The properties of superheated steam tend to approach those of a perfect gas, and the greater the amount of superheating, the greater does this resemblance become. Our present knowledge of steam in this condition is, however, not great.

Moist or wet steam.—If heat be abstracted from saturated steam, the pressure being kept constant, a portion of the steam liquefies, and the steam becomes *supersaturated* or *moist steam*.

Expansion generally.—We will now consider more particularly the subject of expansion, the laws to which air and steam conform during expansion, and the case of steam expanding in the cylinders of a steam-engine. If the vessel or chamber in which any gas is confined be enlarged or contracted, the gas will still completely fill the vessel, but at an altered pressure.

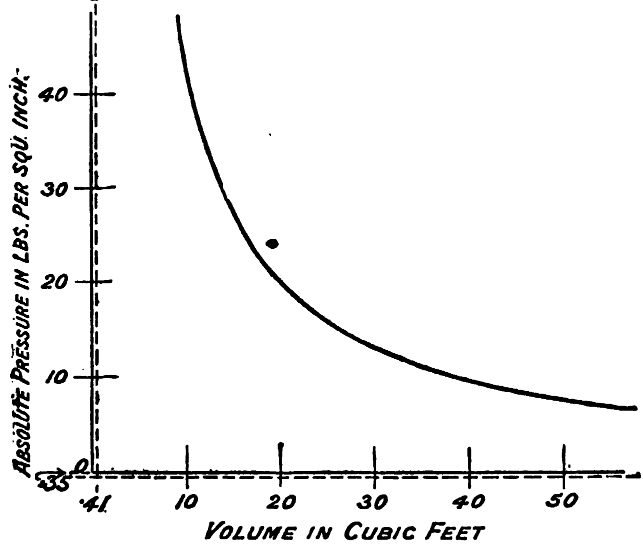


FIG. 128.

Expansion of a perfect gas.—During the process of expansion of a perfect gas, of which atmospheric air may be taken as a type, the pressures and volumes are connected by the law that their product is always proportional to the absolute temperature, or if

p = the pressure,
v = the volume of one pound of the gas,
and T = its absolute temperature,
then $p\ v = c$ T, where c is a constant quantity.

If the temperature remain constant, the alteration of pressure will be in inverse ratio to the alteration of volume. For example, if two cubic feet of air at 10 lbs. pressure were compressed into a volume of 1 cubic foot, and the temperature were unaltered, its pressure

would be increased to 20 lbs. per square inch. If it were allowed to expand into a volume of 4 cubic feet, its pressure would be reduced to 5 lbs. per square inch, and so on. This law is generally expressed by saying that if the temperature remain constant, the pressure varies inversely as the volume, or that the product of the pressure and volume of a perfect gas is constant. Therefore, in this case, $p \times v =$ constant.

Geometrical representation.—This may be shown graphically by means of the ordinates of a rectangular hyperbola referred to its asymptotes as axes, this curve representing the law of expansion of air and other perfect gases; the horizontal distances or abscissæ representing the volumes, and the vertical distances, or ordinates, the pressures. In Fig. 124, O X and O Y are two axes drawn at right angles to each other, O being the origin of co-ordinates. Let P M represent the pressure of the gas when its volume is represented by O M, and Q N the pressure corresponding to the volume O N. Then, by hypothesis, O M $\times$ M P $=$ O N $\times$ N Q $=$ constant, or $p\, v =$ constant. The curve passing through a series of such points will therefore be represented by an equation of the form $x\, y = c$, which is that of a rectangular hyperbola, to which the asymptotes are axes, the ordinates representing pressures and the abscissæ volumes.

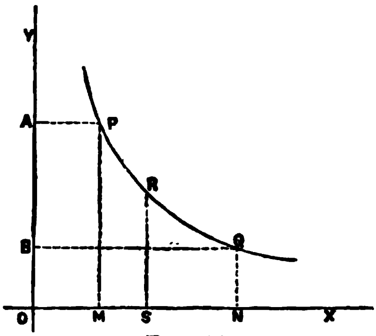


FIG. 124.

The pressure corresponding to any other volume, O S, is found by drawing the vertical ordinate through S, cutting the curve in R, the line R S representing the required pressure. The work done by the gas in expanding from the volume O M to the volume O N is represented by the area of the figure P Q N M.

Expansion of steam.—The laws followed by steam during expansion are different from those just described of a perfect gas, though the general character of the expansion curve is similar.

If $w =$ the work done during expansion
$q =$ the heat added or subtracted during expansion
$i_1 =$ the heat contained in the steam at the beginning of expansion in excess of that contained in the feed water,
$i_2 =$ the similar excess at the end of expansion,

then, by the law of conservation of energy,

$$i_1 + q = w + i_2 \quad \ldots\ldots\ldots\ldots \quad \text{(I)}$$

Further, if the equation to the expansion curve is of the form $pv^n =$ const., then its area bounded by the axis of v, the curve, and two ordinates p_1 and p_2 is given by the expression—

$$w = \frac{p_1 v_1 - p_2 v_2}{n - 1} \quad \ldots\ldots\ldots\ldots \quad \text{(II)}$$

provided that n is not equal to 1. If $n = 1$ the curve is a rectangular hyperbola, and its area is then given by

$$w = p_1 v_1 \log._e \frac{v_2}{v_1}.$$

Free expansion.—There is an important difference to be noted between the free expansion of steam—that is, its expansion without the performance of any mechanical work—and its ordinary expansion in the cylinders of a steam engine, in which during expansion it exerts pressure on the piston and performs work. It is very necessary that this difference should be borne in mind in considering the expansive action of steam in an engine.

Imagine one pound of saturated steam at any given pressure to be confined in a cylinder behind a piston, both the cylinder and piston being made of non-conducting materials. Suppose the piston to be very rapidly moved by an *external* force, so that the volume of the steam is increased without its having performed any work on the piston. Then it is evident that as no heat has been either added to or subtracted from, the steam during the process, the total amount of heat in it is the same at the end of the expansion as it was at the beginning. But, as was pointed out in Chapter III., the total heat of saturated steam increases slowly with its pressure, so that, since the steam was saturated at its original pressure, the total amount of heat in the steam is more than sufficient to keep it in a state of saturation at the reduced pressure at the end of the expansion, so that the steam will be to some extent superheated.

To take a numerical example, suppose one pound of saturated steam at a pressure of two atmospheres to be allowed to expand in a perfectly non-conducting vessel, without doing any mechanical work, to a pressure of one atmosphere. Since the steam expands without doing any external work, and without the addition or subtraction of any heat, both w and q in equation (I) (page 147) are equal to zero, so that

$$i_1 = i_2$$

also the steam being dry and saturated at the commencement of the expansion—

$$\begin{aligned} i_1 = \mathrm{I}_1 &= t_1 - t_0 + 966 - \cdot 7\,(t_1 - 212) - \cdot 1851\, p_1 \mathrm{V}_1 \\ &= 249 - 32 + 966 - \cdot 7\,(249 - 212) - \cdot 1851 \times 29\cdot 4 \times 13\cdot 7 \\ &= 1082\cdot 6. \end{aligned}$$

$$\begin{aligned} \text{Also } i_2 &= t_2 - t_0 + x\,\{966 - \cdot 7\,(t_2 - 212) - \cdot 1851\, p_2 \mathrm{V}_2\} \\ &= 212 - 32 + x\,\{966 - \cdot 7\,(212 - 212) - \cdot 1851 \times 14\cdot 7 \times 26\cdot 3\} \\ &= 180 + 894\cdot 2x, \text{ and hence} \end{aligned}$$

$$1082\cdot 6 = 180 + 894\cdot 2x$$

$$\text{and } x = 1\cdot 009$$

a quantity greater than 1, so that the result of expanding the steam without doing external work is to superheat it. (See page 31.)

The phenomenon of *free expansion*, or expansion without the performance of mechanical werk, is one that often occurs in triple expansion and other stage expansion engines on the admission of steam to the receivers or reservoirs between the cylinders. Taking a triple expansion engine, the final pressure in the high-pressure cylinder is generally somewhat higher, and in many cases considerably higher, than the initial pressure in the intermediate cylinder, so that when the steam escapes from the high-pressure cylinder to the intermediate receiver, its volume is suddenly increased without any external work being done; and the difference between the amount of heat in the

steam at the end of the stroke in the high-pressure cylinder and that necessary to keep it in a state of saturation at the reduced pressure of the receiver is expended in superheating it, if it were saturated on release, or in drying it, if, as usual, it be moist on release.

Work done during expansion.—When, however, steam during its expansion *performs mechanical work*, the conditions of the case are very different from those just discussed. We will in the first place assume that the expansion takes place in a perfectly non-conducting cylinder, so that heat is neither added to nor subtracted from the steam during the operation. Until the true nature of heat was first determined, it had been supposed that when steam was expanded in this way, the total amount of heat in it was the same at the end as it was at the beginning of the expansion. It was, however, always found that water collected in the cylinders, but this was supposed to be due to priming, or the carrying of spray from the boilers to the cylinders, which explanation was often found to be unsatisfactory.

The real cause of the presence of this water in the cylinders was, however, readily explained when the principles of thermo-dynamics became understood. It was then seen that the mechanical work done by the steam during the expansion was due to the fact that a portion of the energy that had been stored in the steam in the form of heat had become transformed into mechanical work, and appeared no longer in the form of heat, so that the total quantity of heat contained in the steam had been diminished. The abstraction of the amount of heat thus changed into mechanical work was sufficient not only to lower the temperature of the steam to that corresponding to its reduced pressure, but also to cause some of it to liquefy and become water.

For example, suppose one pound of *saturated* steam at an absolute pressure of 60 lbs. per square inch, to expand in a non-conducting cylinder, without addition or subtraction of heat,[1] pressing a piston before it, till its pressure fell to $3\frac{1}{2}$ lbs. per square inch absolute,

In this case since no heat is added or subtracted $q = 0$, so that equation (I.), page 147, becomes $i_1 = w + i_2$

$$\therefore w = i_1 - i_2 \quad . \quad . \quad . \quad . \quad \text{(III.)}$$

also as the steam is dry and saturated at the commencement of expansion we can write (see page 31)

$$i_1 = \mathrm{I}_1 = t_1 - t_0 + 966 - \cdot 7(t_1 - 212) - \cdot 1851 p_1 \mathrm{V}_1.$$

The condition of the steam, as regards wetness, at the end of expansion, will for the present be assumed unknown, the dryness fraction being called x, so that we write

$$i_2 = t_2 - t_0 + x\{966 - \cdot 7(t_2 - 212) - \cdot 1851 p_2 \mathrm{V}_2\}$$

Now $t_1 = 292{\cdot}5$ and $t_2 = 147{\cdot}69$. Also V_1 the specific volume of steam at 60 lbs. pressure absolute $= 7{\cdot}037$ cubic feet and V_2 the same for $3\frac{1}{2}$ lbs. pressure $= 101{\cdot}665$. Substituting these values we get

$i_1 = 1092$ and $i_2 = 115{\cdot}69 + 945{\cdot}15x$ so that $i_1 - i_2 = 976{\cdot}31 - 945{\cdot}15x$.

From equation II., page 147

$$w = \frac{\mathrm{P}_1 \mathrm{V}_1 - \mathrm{P}_2 v_2}{n - 1} \quad . \quad . \quad . \quad . \quad \text{(IV.)}$$

[1] Termed adiabatic expansion. (See later in this chapter.)

the expansion being adiabatic the equation to the curve becomes

$$pv^{1\cdot13} = \text{constant (see page 153)}$$

$$\text{so that } p_1 V_1^{1\cdot13} = p_2 v_2^{1\cdot13}$$

and substituting the known values for $p_1 V_1$ and p_2 we obtain

$$v_2 = 86\cdot99.$$

From equation (IV.)

$$w = \frac{144(p_1 V_1 - p_2 v_2)}{\cdot 13}$$

$$= 130{,}420 \text{ ft. lbs.}$$

$$= 167\cdot54 \text{ British thermal units.}$$

and substituting in equation (III.)

$$167\cdot54 = 976\cdot31 - 945\cdot15x$$

therefore $\quad x = \cdot8557$

from this it is seen that $1 - x = \cdot1443$ of the pound of steam has been condensed in order to provide the heat equivalent of the work done.

The volume of steam at the end of the adiabatic expansion has been shown to be equal to 86·99 cubic feet. The volume of saturated steam at the same pressure is equal to 101·665, the dryness fraction x is therefore equal to $\frac{86\cdot99}{101\cdot665} = \cdot8557$, the same value as obtained above.

Liquefaction in cylinders.—In the preceding case, in which the steam has been supposed to expand in a non-conducting cylinder, the water of liquefaction would simply be carried to the condenser at the end of each stroke and no *waste* of heat would ensue. Unfortunately, however, we have to deal in practice with very different conditions, as the cylinders and pistons are necessarily made of conducting materials, by which we shall see the loss from liquefaction in the cylinders becomes very great.

Considering a low-pressure cylinder for the sake of illustration, the hot steam enters the cylinder after it has been open to the condenser for a whole stroke, and when the temperature of the metal to a certain depth below its internal surface may be supposed to approximate to that of the steam passing to the condenser, say from 140° to 150° Fahr. It is therefore evident that a quantity of the entering steam will be condensed on these cool surfaces, and the heat given up by this condensed steam will be expended in raising the temperature of the cylinder, cylinder cover, piston, &c.

As the steam expands, a further portion of it liquefies, due to the work done, and probably exists in the form of spray, or collects on the surfaces of the cylinders, &c. Consequently, when the pressure of the steam has fallen, and its temperature is below that of the metal of the cylinder with its film of condensed steam, this film of water immediately commences to evaporate, as its temperature is higher than that due to the pressure of steam in contact with it, and it will evaporate till its temperature corresponds to that of the expanding steam. It also abstracts heat from the metallic surfaces, tending also to reduce them to the same temperature.

During the period of exhaust, when the steam pressure has fallen to, say, 2 to 3 lbs. absolute, under which pressure water boils at about 140° to 150° Fahr., evaporation of the film of water becomes much more

rapid, and further heat is also abstracted from the metallic surfaces. The heat abstracted originally from the entering steam by this agency goes direct to the condenser, and not only does no useful work, but increases the back pressure on the piston to an extent sensibly felt in many unjacketed engines. The cylinder being now in this comparatively cool state, fresh steam enters, and condensation again takes place, with deposition of a film of water, and again raising the temperature of the cylinder surfaces.

This process goes on at every stroke, and the great loss that generally arises from liquefaction in the cylinders of a steam-engine is therefore due to the fact that the water in the cylinder, existing probably as a film on the surfaces, acts as an equaliser of temperature, lowering the initial and raising the final temperatures and pressures, and thus decreasing the efficiency of the steam. The effect is the same as if, during each stroke, a certain portion of the steam passed direct from the boiler to the condenser, without performing any work whatever.

The action of any water remaining in the clearance spaces and pockets is also similar, and also occasions a direct transfer of heat to the exhaust.

Experiments on liquefaction at various rates of expansion in simple engines with unjacketed cylinders.—In some experiments made by Mr. Isherwood, of the United States Navy, it was found that with an expansion of only four times, the amount of steam wasted as described above was more than that performing work, so that the expenditure of heat was more than doubled. The experiments were made on simple engines with unjacketed cylinders, having a piston speed of about 224 feet per minute supplied with saturated, or perhaps rather moist steam. Comparing the actual water used, by measurement, with that shown by the indicator diagrams, the results were that the amount of steam wasted by condensation, clearance, and leakage was 15 per cent. with a ratio of expansion of 1·07, rising to 46 per cent. with an expansion of $2\frac{1}{2}$ times, and to 61 per cent. with a ratio of expansion of 4.

In the United States vessel 'Michigan,' with a steam pressure of 20 lbs., the least consumption of steam per I.H.P. per hour was 32·67 lbs. with a cut-off $\frac{1}{3}$; it did not differ much for cut-offs between $\frac{1}{4}$ and $\frac{7}{10}$, while with greater expansion the consumption rapidly increased, till with a cut-off of about $\frac{1}{11}$ it had risen to 46 lbs. per I.H.P. Mr. Isherwood inferred from these trials, having regard to the effect on the size of the cylinders, that the best point of cut-off for naval vessels of this type was about $\frac{7}{10}$, the consumption then being 34·8 lbs. per I.H.P. per hour.

Cutting off earlier than at $\frac{1}{3}$ of the stroke resulted in loss instead of gain, so that we see how serious was the limitation imposed by practical considerations on the attainment of the theoretical advantage due to expansion.

Owing to the low piston speed and small size of the engine, these cases were unfavourable as regards liquefaction, as both items are powerful factors in increasing the losses due to this, but the experiments, however, show how great the losses by liquefaction become when no provision is made to prevent or reduce them.

As showing the influence of steam pressure on these results another experiment by Mr. Isherwood, on a vessel using steam at the higher

pressures of from 40 to 50 lbs., may be mentioned. In this case the most economical point of cut-off was at ·38 stroke, 30·3 lbs. of steam being used per I.H.P. per hour, but there was little difference in economy up to a cut-off of ·56 stroke, which gave a consumption of 30·62 lbs. per I.H.P. At an earlier cut-off than ·38 there was again a loss. With the higher pressure, therefore, we see that expansion could be carried economically to a greater extent.

At still higher pressures the loss from liquefaction remains very considerable unless its effects be counteracted.

Non-conducting materials required for efficient expansion.—The highest theoretical efficiency in the expansive working of the steam can only be realised if the cylinders and pistons are made of perfectly non-conducting materials. It is not sufficient to cover the exterior of the cylinders with such non-conducting materials, which only prevent the passage of heat from the steam to the atmosphere and not the complex action which goes on in the cylinder by the abstraction of heat from the steam during the admission, which heat is again given out to the steam during the exhaust. In this process it is only necessary that the metal of the cylinder should be cooled for a very small distance below the surface, which is probably what happens in practice.

Influence of size of engine.—Small engines with unjacketed cylinders are less economical than large ones. This is easily seen when we consider that the smaller the diameter of cylinder, the greater is the ratio of the surface of the cylinder, &c., which is alternately heated and cooled, to the volume of steam contained in the cylinder, so that the amount of liquefaction is proportionately increased, and the quantity of heat taken up by the metal of the cylinder during admission and given out during exhaust will be proportionately greater.

To take a simple case for illustration. In a cylinder one foot in diameter and one foot long the area of the cylinder surface is 3·1416 square feet, and the area of the piston and cylinder cover 1·5708 square feet, making a total surface of 4·7124 square feet. The volume of the cylinder is ·7854 cubic foot, so that the ratio of surface to volume is 6 to 1. In other words, there are six square feet of heating and cooling surface to one cubic foot of steam used. Now suppose the diameter of the cylinder to be doubled, the stroke remaining the same. In this case the volume is increased in the ratio of 4 to 1, the cylinder containing 3·1416 cubic feet of steam. The areas of the piston and cylinder cover are also increased in the ratio of 4 to 1, but the internal surface of the cylinder is only increased in the ratio of 2 to 1, and the total surface is 12·5664 square feet. In this case, then, there are only 4 square feet of surface for each cubic foot of steam instead of 6 square feet, as in the previous example. We conclude, therefore, that with unjacketed cylinders the percentage of loss from liquefaction will be less in large engines than in small ones, and consequently that large engines are more economical than small ones per unit of power developed, when the cylinders are unjacketed.

Adiabatic expansion of steam.—The law which steam follows when expanded in a non-conducting cylinder *without gain or loss of heat* is represented approximately by the equation

$$p\,v^{\frac{10}{9}} = \text{constant}.$$

The curve representing this is called the *adiabatic curve*, and for the same point of cut-off it falls considerably below the hyperbola.

Zeuner gives the following equation to represent it, viz. :—

$$p\,v^{\left(1{\cdot}035 + \frac{x}{10}\right)} = \text{constant}.$$

where x is the initial dryness fraction of the steam.

The smaller x is—that is, the more moisture the steam contains—the nearer will the expansion curve of the steam approximate to the hyperbola.

When the steam is quite dry $x = 1$, and the equation to the curve becomes

$$p\,v^{1{\cdot}135} = \text{constant},$$

which does not differ much from the previous formula

$$p\,v^{\frac{10}{9}} = \text{constant}.$$

Zeuner's equation does not hold good for cases in which the steam contains more than 30 per cent. of moisture, i.e. for values of x below ·7.

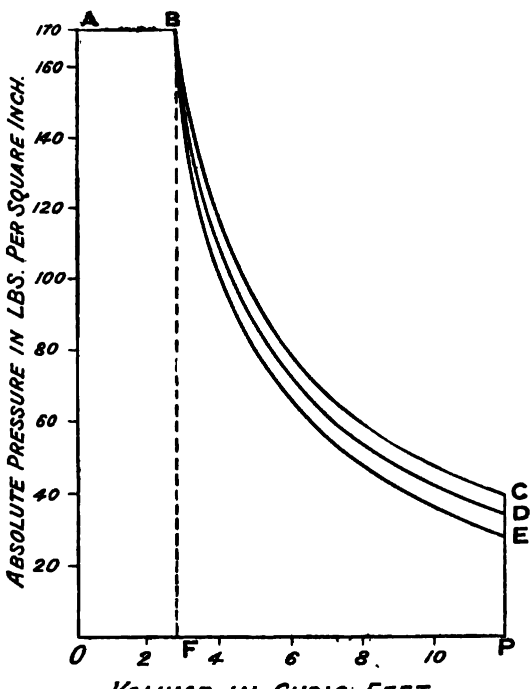


Fig. 125.

Comparison of various expansion curves of steam.—It will be interesting to compare the three expansion curves we have now referred to—viz. the hyperbola, the saturation curve, and the adiabatic curve, representing the expansion of the volume A B of steam to the final volume O P. The three curves will be relatively as shown in Fig. 125.

B C, the hyperbola, lies above the others, B D is the saturation curve, and B E, the lowest, is the adiabatic curve.

As we saw previously, a considerable amount of steam must be liquefied when steam expands adiabatically doing external work. This liquefaction of a certain volume of steam reduces the pressure as the expansion proceeds, which explains the falling away of the curve B E from the curve B D, which represents always the same weight of dry saturated steam. The difference D E represents the final fall of pressure.

If steam expands according to the saturation curve B D, then clearly heat must be added to the expanding steam by a steam jacket or other means to bring about this result. Since the steam is dry and saturated, i_1 and i_2 in equation (I.) page 147, become I_1 and I_2 so that we can write the equation

$$q = w - (I_1 - I_2)$$

so that the amount of heat to be added is less than the external work done by the difference in the heat contained in the steam at the beginning and end of expansion.

If the amount of heat added to the steam when expanding exceeds that referred to above for the curve B D, the steam will become superheated if it be initially dry, but the transfer to the steam of such an amount of heat as to superheat it, is very improbable in practice. If the steam initially contains a certain amount of water, which is the usual condition in practice, a portion of the moisture will be evaporated. In each case a rise of pressure ensues beyond the saturation curve, while if the supply of heat is sufficient, the curve will approximate to the hyperbola B C, which represents the expansion of a perfect gas at a constant temperature.

The amount of heat necessary to be added to steam during expansion to maintain the hyperbolic curve can be approximated to as follows:—Since the expansion is hyperbolic the work done during expansion

$$w = \cdot 1851 p_1 V_1 \log_e r \text{ (page 147)}$$

the steam being supposed dry at the commencement of expansion $i_1 = I_1$.

The hyperbolic curve B C falling above the saturation curve B D, the volume of the steam for any given pressure is greater than for saturated steam and therefore the steam must be superheated so that i_2 is greater than I_2.

Equation (I.) page 147, therefore becomes

$$q = \cdot 1851 p_1 V_1 \log_e r - (I_1 - i_2)$$

where i_2 is greater than I_2, so that q is greater than $\cdot 1851 p_1 V_1 \log_e r - (I_1 - I_2)$ but by how much cannot be accurately calculated.

It will be found that considerable differences in the amount of heat supplied will make very small differences in the form of the expansion curve, and this renders it difficult to analyse the performances of engines from the indicator diagram. The result is, that for practical purposes the expansion curve can be assumed to be the ordinary hyperbola, by which calculations of pressure and power are facilitated.

Construction of various curves of expansion.—The hyperbolic curve is easily drawn by the geometrical rule previously described. The saturation curve can be most readily drawn from a table of properties of steam, as given in Chapter III. The adiabatic curve is constructed by calculation.

The analysis of the numerous questions affecting the action of steam during expansion is further facilitated by the use of the 'temperature-entropy' method in lieu of the 'pressure-volume' method described.

CHAPTER XIII.

METHODS OF INCREASING THE EXPANSIVE EFFICIENCY OF STEAM.

THE methods which are or have been adopted for preventing or reducing the loss by the liquefaction of steam in the cylinders of a steam-engine, and thus increasing its expansive efficiency, are :—

1. Surrounding the cylinder with a casing or jacket kept full of steam of high temperature, i.e. 'steam-jacketing.'
2. Superheating the steam before it is admitted to the cylinder.
3. Dividing the expansion of steam into stages, as in compound engines (either double, triple, or quadruple expansion).
4. By the use of rotary engines, in which useful expansion can be carried much further than with reciprocating engines, the number of stages of expansion being much greater.

Steam-jacketing.—The steam jacket was invented by Watt, but it is not certain that he properly understood the principles of its action, and most engineers of that period, arguing from the erroneous theory of caloric, then generally accepted, deemed it unnecessary, discontinued its use, and considered it sufficient to clothe the cylinders carefully with non-conducting materials to prevent loss from radiation.

The use of the steam jacket was, however, retained in a few special cases, such as the pumping engines for the Cornish mines, and these engines were for many years famous for their economy.

In almost all other engines of the period under review, and certainly in all marine engines, steam jackets were not fitted. The result was that little or no practical advantage ensued if the steam were expanded more than from two to three times, and the truth of this was manifested not only by experiments, but also by everyday experience in the working of engines.

When, however, the true nature of the action of expansion in a steam cylinder was discovered, and it was ascertained that the work of the engine was performed by the abstraction of heat from the steam and its conversion into mechanical work during expansion, which caused a portion of the steam to liquefy, it was seen to be necessary, in order to increase the efficiency of the steam, to provide for the addition of heat to the steam during expansion, and the reintroduction of steam jackets has taken place, and they are now fitted to most modern engines.

Action of the steam jacket.—The effect of the jacket is to reduce the changes of temperature of the metal of the cylinder that take place in the unjacketed engine. The heat added to the steam expanding in the cylinder should be just sufficient to prevent any appreciable quantity of it becoming liquid, and under these conditions the expansion

diagram should be a curve representing the successive pressures and corresponding volumes of a given weight of saturated steam.

The work done by the steam necessarily causes liquefaction to take place somewhere in jacketed as well as unjacketed engines; but in the former case, if the steam jacket is able to supply sufficient heat to the expanding steam, this liquefaction takes place in the jacket, where it produces no subsequent bad effect, and the condensed steam is simply collected and returned again to the boiler, and no *waste* of heat ensues in consequence, as pointed out in Chapter XII.

The preceding remarks explain the theory of the economy due to steam jackets, but in practical cases liquefaction is not prevented entirely. On the contrary, even with jackets, there is usually considerable liquefaction, but it is reduced by jacketing.

Moisture in steam considerably increases its power of conduction of heat. By means of the steam jacket the steam is kept drier, so as to be a bad conductor of heat, and the moisture it contains, though probably sufficient to lubricate the piston, is thus prevented from increasing to such an extent as to carry away any considerable amounts of heat from the metal of the cylinder and piston to the condenser.

Extent of steam-jacketing.—Steam jackets were at first fitted to the barrels of cylinders only; they were then added to the covers and ends, and in some special cases complicated arrangements were made, by fitting hollow piston-rods and telescopic steam pipes, to admit steam to the interior of the piston. There is, however, more advantage to be derived from jacketing the barrels than from jacketing the ends or the piston, because the friction of the piston keeps the surface of the cylinder barrel comparatively clean, while the surfaces of the ends and piston soon become covered with a deposit which interferes with the passage of heat. Pistons are now never jacketed and the ends and covers are also generally unjacketed. In the Royal Navy only the cylinder barrels are steam-jacketed in modern vessels.

Experiments to prove the economy due to steam-jacketing.—Many such experiments have been made. The late Mr. John Penn made some, but although the working pressure was only 7 lbs. per square inch, when the cylinders were jacketed with steam of this pressure, the gain was considerable and the economy increased as the pressure in the jacket was increased.

Some valuable experiments to ascertain the efficiency of steam jackets were made by Mr. Emery on the cylinders of the 'Bache.'

When this engine was worked as a simple expansion engine, with steam pressure of 80 lbs., the most economical results were obtained, both with and without the jacket in use, when the ratio of expansion was about five times. Above this amount of expansion the consumption of steam increased considerably even when the jackets were used, while without the jackets in use the consumption increased with much greater rapidity. When the jacket was not in operation 26·25 lbs. of feed-water were required per I.H.P. per hour, whilst when the jacket was used only 23·15 lbs. were required. At the higher rates of expansion the percentage of saving due to the jacket was greater.

When the engine was worked as a compound engine, the amount of feed-water required per I.H.P. per hour was practically the same from a total rate of expansion of 5·7 up to a total rate of 9·2, the most

economical results being obtained with between 6 and 7 expansions. Below 5·7 expansions there was a loss, as compared with 6 or 7 expansions, while at 16·8 expansions there was a considerable loss. For about the same expansion of $6\frac{1}{2}$ to 7, the consumption of feed-water when the jacket was not used was 23·03 lbs., whilst when the jacket was used only 20·33 lbs. were required per I.H.P per hour.

Experiments of the Institution of Mechanical Engineers, &c.—The Institution has made valuable inquiries on this question, showing that from 10 per cent. to 17 per cent. was saved by the steam jacket in double or triple expansion engines, and about 20 per cent. in simple engines. In many cases a comparatively small quantity of steam liquefied in the jacket caused a very large saving of total steam. In one experiment on a small engine, a condensation in the jackets of 7 per cent. of total steam effected a saving of 25 per cent. in total steam.

Professor O. Reynolds has also recorded experiments made with a small triple expansion engine, showing that the liquefaction amounted to 40 per cent., even when the expansion was split up into three stages, and that the steam was rendered nearly dry on exhausting to the condenser by jacketing all cylinders with boiler pressure steam.

On the other hand, experiments made in H.M.S. 'Argonaut,' with and without steam in jackets, must be mentioned. These showed but little difference between the two methods; in fact, the water consumption with steam in jackets was slightly greater than without steam, but the jacket pressures used did not much exceed the initial pressure in the cylinders, while the engines are large and of high speed. Similar results have been obtained in other recent ships.

Jacket steam pressure.—From the explanation given of the action of steam jackets, it appears that for high efficiency they should, if practicable, be filled with steam of a considerably higher temperature than that being admitted to the cylinder, for as there must be some difference in temperature between the inside and the outside of the cylinder to cause heat to flow to the inside surface, the temperature of the inside surface of the cylinder would be otherwise less than that of the entering steam. Unfortunately, however, practical difficulties are met with if the jacket pressure is high compared with the initial pressure, and several cases of scoring of cylinder liners in the Royal Navy have been attributed to this cause. It seems undesirable to pass the steam, *on its way* to the cylinder, through the jacket, because in this case it would be partially condensed before admission to the cylinder, and its efficiency reduced by the presence of water.

Influence of size on economy.—Calculation shows that the jacket area is comparatively less in large than in small engines, for whilst the volumes of the cylinders increase as the square of the diameter, the area of the jacket surface only increases directly as the diameter. The percentage of saving from the use of the jacket may, therefore, reasonably be expected to be greater in small than in large engines.

Influence of speed.—Similarly, as regards speed, it will be readily admitted that with low speeds the cylinder will have sufficient time in which to abstract heat from the steam and give it up again during exhaust. When, however, the speed becomes very great, the changes between steam and exhaust are so rapid that there is not sufficient time for the cylinder walls to exercise their full deleterious effect. This

tion temperature. In the writer's opinion, superheating to this extent is quite possible, for although difficulties exist, they are probably not now insurmountable, and the practice is worthy of further trial.

The use of rotary engines.—With rotary engines, such as the steam turbine, steam of certain pressure is always in contact with the same portion of the cylinder, and there are no variations of temperature causing re-evaporation of liquefied steam as in reciprocating engines, so that this cause of inefficiency is absent. This facilitates the use of a high expansion with efficiency, while the provision of the requisite volume for this expansion does not result in such losses due to friction of large moving parts as when this is attempted in a reciprocating engine. A turbine can therefore deal economically with a very low pressure of steam, or in other words a very high vacuum.

The following examples have been given by Mr. Parsons from tests made on land engines, showing the effect of high vacuum upon the steam consumption of reciprocating engines and turbines. Fig. 125*a* shows the effect of vacuum on the consumption of a reciprocating

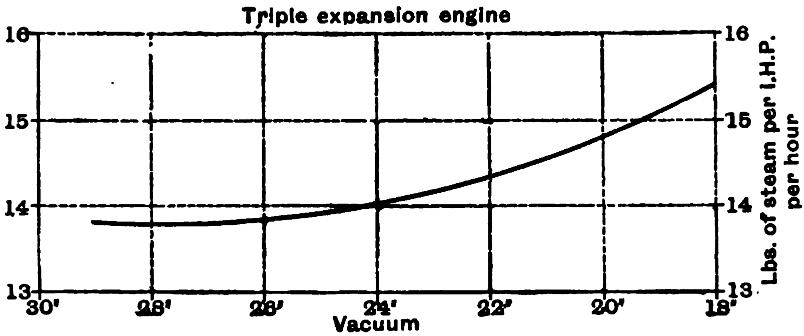


FIG. 125*a*.

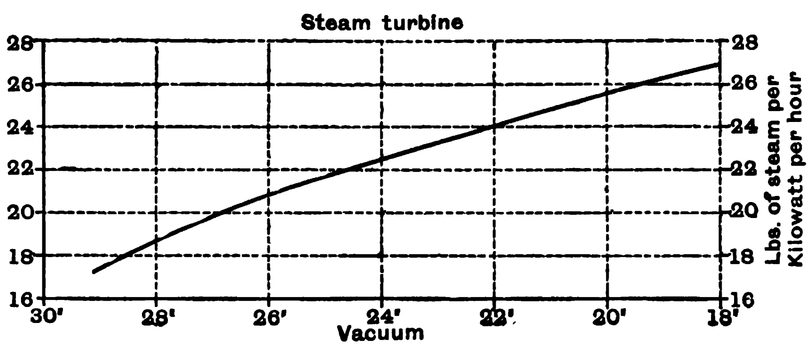


FIG. 125*b*.

engine of 200 I.H.P., while Fig. 125*b* shows its effect upon the steam consumption of a steam turbine driving a 1,000 kilowatt dynamo.

Fig. 125*a* shows that in the reciprocating engine the consumption

falls with reduction of vacuum down to 25 or 26 inches, after which there is little further gain. On the contrary, if Fig. 125*b* be studied, the result appears that with the turbine the consumption continues to fall fairly rapidly much beyond the vacuum of 25 or 26 inches.

Superheating with turbines.—A considerable gain also results from the use of superheated steam with turbines, but although the same difficulties as with reciprocating engines are not met with, others take their place, such as the increased expansion of parts and interference with clearance due to the higher temperature of superheated steam. The theoretical effect of 50° F. of superheat at several receiver pressures is indicated by dotted lines 0_s, 50_s, 100_s, etc., in Fig. 125*c*, which shows

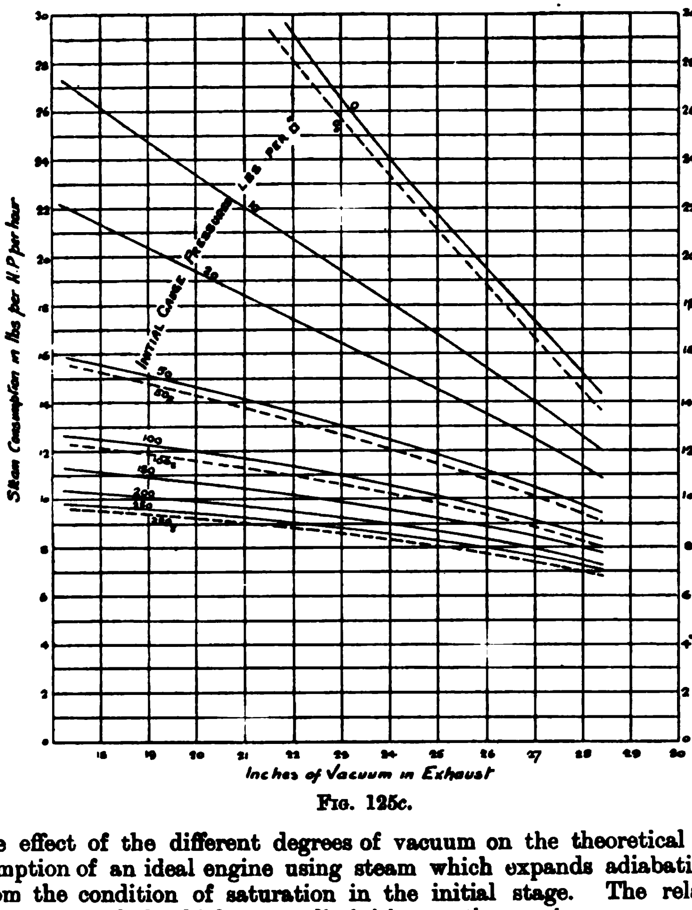


Fig. 125*c*.

the effect of the different degrees of vacuum on the theoretical consumption of an ideal engine using steam which expands adiabatically from the condition of saturation in the initial stage. The relative importance of the high vacua diminishes as the receiver pressures are increased, and in considering the percentage gains due to particular increases of vacua it is essential that the initial and final conditions of the steam be considered.

In practice the steam does not expand adiabatically owing to internal losses due to tip clearance, blade and eddy friction, etc., and in general the effects of vacuum are from this reason probably not so pronounced.

CHAPTER XIV.

COMPOUND OR STAGE EXPANSION ENGINES.

We have referred to steam-jacketing, and in a lesser degree to superheating, as tending to prevent liquefaction in engine cylinders, but they are far from doing so entirely, in ordinary engines, and in order to realise the full benefits of a high ratio of expansion, the system of dividing the expansion into stages, carried out in two or more separate and successive cylinders, must be adopted. Engines of this description are generally called '*compound or stage expansion engines.*'

Invention and abandonment of compound engines.—This system was invented as far back as 1781, but was, however, soon abandoned, for it is principally adapted for high pressures, which were not then in use; but when the fact was fully accepted that in order to make long voyages remunerative, the pressures and rates of expansion of steam must be increased so as to reduce the expenditure of coal, the question of the stresses brought on the framing and shafting of the engine by working the steam at a high rate of expansion in a single cylinder became one of great importance; as, in large engines especially, the variation of pressure during the stroke would be so great that the maximum stresses produced would probably be dangerous to the structure unless it were made excessively strong. Attention was again directed to the employment of the compound engine, in which the high-pressure steam acts on a small piston only, and a reduced pressure on the large piston, which reduces the maximum stresses on the framing, &c., and makes the turning moments more uniform.

Re-introduction of system. Causes of advantages.—With the increase of pressure the system was re-introduced, and the economy resulting was so decided that its application for marine purposes soon became universal. As the working pressures increased additional stages in the expansion became desirable, and this led to the *triple* and *quadruple expansion* engines now so extensively used. The principle is simply an extension of James Watt's idea of keeping the steam vessel or cylinder as warm as possible and the condenser as cool as possible. With simple condensing engines, the cylinder into which the boiler steam is admitted is also open to the condenser for nearly the whole period of the return stroke of the piston, so that its temperature, or at least that of a certain layer of thickness of the internal surface, together with any water remaining in the cylinder, may be supposed to be considerably cooled during this part of the stroke, to be again raised in temperature by the liquefaction of the entering steam, thereby causing a considerable loss due to the direct

transfer of heat to the condenser, without the performance of any work, as previously explained.

This loss by liquefaction is greater, the greater is the difference between the initial and final temperatures, so that with increased pressures and temperatures, the difference or range of temperature in the cylinder becomes greater, and the loss from this cause when expanding to the full extent in a single cylinder is proportionately increased. If, however, we divide the expansion into two or more stages, the cylinder into which the high-pressure steam is admitted is never open to the condenser, and its temperature is never reduced below that of the intermediate receiver; also, the steam condensed and evaporated in the first cylinder re-appears as working steam in the second cylinder, instead of passing straight to the condenser. The useful work done by it is one source of the economy of stage expansion engines. The loss from liquefaction in the second cylinder is also reduced, in consequence of the smaller range of temperature between admission and exhaust in that cylinder.

Another way in which the adoption of stage expansion engines has increased the efficiency of the steam, is by reducing the clearance spaces into which the boiler steam is admitted. These clearance spaces are much smaller in the high-pressure cylinder of a compound engine than they would be had the whole expansion taken place in one large cylinder. At each stroke of the engine this space has to be filled, while no work is being done by the piston, so that the loss of efficiency due to the waste of steam by clearance spaces is much less in the compound engine than in the simple engine. On the other hand, the compound or stage expansion engine has losses of efficiency, due to sudden expansion and wire-drawing between the cylinders, which do not exist in the simple engine, but these losses are of much smaller magnitude than the gains just described, so that to obtain the highest economy from high-pressure steam the stage expansion engine is essential.

We have only referred in this chapter to the effect on the efficiency of steam by the use of compound or stage expansion engines, but, as will appear later, this type of engine has other advantages.

Trials of double compound versus simple engines with the same steam pressure and ratio of expansion.—The experiments previously referred to, made by Mr. Emery on the engines of the 'Bache,' gave a great deal of valuable information as to the comparative efficiencies of the two systems for the utilisation of steam.

The results were shortly that the consumption of water per I.H.P. per hour was always considerably less in the compound engine than in the simple expansion engine when these were working at about the same rate of expansion. This was shown to be the case when the cylinders were jacketed, as well as when they were not jacketed.

With steam jacket in use and with an expansion of between five and six times, which proved to be the most economical rate for each type of engine, the feed-water used per hour was in the simple engine 23·15 lbs. per I.H.P., whilst in the compound engine it was only 20·36 lbs. per I.H.P., showing in this case a gain in economy at 80 lbs. steam

pressure by the use of the compound engine of rather over 12 per cent. At the higher rates of expansion the percentage of economy due to the compound engines was still higher.

The consumption of feed-water for the power being developed was, both in the compound engine and in the simple engine, considerably greater at the higher rates of expansion than at the lower rates.

In each case, therefore, the efficiency of the steam was considerably reduced when the rate of expansion was increased beyond a certain point.

Tri-compound or triple expansion engines.—For steam pressures above 120 lbs. to 130 lbs. per square inch, which are now generally used, it has been found desirable to extend the compound system by dividing the expansion into three stages, so as to reduce the range of temperature in each cylinder, and still further limit the effects of liquefaction. The triple expansion type is now the most common one for modern marine engines, and the gain in economy by its use over the previous double compound engines fitted is well established. The saving of fuel with a triple expansion engine of 150 lbs. to 160 lbs. pressure may be taken as about 20 per cent. compared with the compound engine of about 90 lbs. pressure.

The consumption of good fuel with the most successful triple expansion engines, where the engines are designed principally with a view to economy only, as in many vessels of the mercantile marine, is reported to be from $1\frac{1}{3}$ lbs. to $1\frac{1}{2}$ lbs. per I.H.P. per hour. The average of 28 steamers collected by Mr. Blechynden in 1891 gave 1·52 lbs. as the average consumption per I.H.P. on trial, with steam pressure averaging 158 lbs. per square inch at 64 revolutions per minute. The average consumption on long sea voyages in vessels built at that period appears to be about 1·75 lbs. The average of a number of vessels built about

—	Double compound, cylinders not jacketed			Triple compound, more or less jacketed. (See line 9)		
Name of vessel . . .	Fusi Yama 1	Colchester 2	Ville de Douvres 3	Meteor 4	Tartar 5	Iona 6
1. Duration of trial hrs.	14	11	9	17	10	16
2. Pressure in boilers lbs.	71	95	120	160	158	179
3. Total ratio of expansion	6·1	6·1	5·7	10·6	15·7	19
4. Condenser vacuum ins.	25	25	$20\frac{1}{2}$	$24\frac{1}{2}$	26	$27\frac{3}{4}$
5. Revolutions per minute.	55·6	86·5	86·8	71·8	70	61·1
6. I.H.P.	371	990	2977	1994	1087	645
7. Actual water used per I.H.P. per hour by measurement } lbs.	21·17	21·78	20·77	14·98	19·83[1]	18·35
8. Percentage of water used, accounted for by indicator diagrams of L.P. cylinder	70·8	52·7	72·5	75·8	60·8	59·1
9. Particulars of cylinder jackets	None	None	None	All	I. & L. only	H. only

[1] Probably included priming water.

1901, collected by Mr. McKechnie, gave a steam pressure of 180 lbs. with a trial trip consumption of about 1·48 lbs. per I.H.P., and a consumption on long sea voyages of about 1·6 lbs. per I.H.P. The average revolutions had risen to about 87 per minute.

Experiments on double versus triple compound engines.—A valuable series of experiments was made on six steamships by a committee of the Institute of Mechanical Engineers which reported in 1892. A few particulars of these trials, which included double and triple compound engines, are shown on the previous page. The double compound engines were not steam jacketed.

The great efficiency of the 'Iona' should be noticed, due to her high steam pressure of 180 lbs. and expansion of 19 times. Her consumption was 13·35 lbs. of water per I.H.P. per hour.

The conclusion to be drawn from these interesting experiments is that a substantial gain in economy is obtained by the triple expansion engines with higher steam pressures, which is shown by the consumptions of feed-water for the various vessels indicated on line 7.

Quadruple expansion engines.—A still further gain in economy is effected in the mercantile marine by splitting the expansion into four stages, associated with an increased steam pressure and a greater ratio of expansion. The average steam pressure in such engines recently built is 215 lbs., and the coal consumption is reported in some cases to be about 1¼ lbs. per I.H.P. per hour.

General conclusions.—It will appear from the observations made in the preceding chapters :—

1. That economy is increased by the use of higher steam pressures, and expanding the steam, provided the expansion is not excessive.

2. That the amount of expansion required with high-pressure steam can be carried out most efficiently and economically in stage expansion engines, so that the variations of pressure and temperature in each cylinder are comparatively small.

3. That with high pressures and ratios of expansion, surrounding the cylinder with a jacket filled with steam of high temperature adds to the efficiency of the expansion in small and slow-moving engines, but as size and speed are increased there is but little if any gain by the steam jacket.

4. That additional efficiency of the steam would result from the use of superheaters, so that renewed efforts to overcome the practical difficulties attending their use appear desirable.

5. That there is a considerable gain in steam efficiency from the use of rotary engines in the form of steam turbines, expansion being economically carried out in this type to a much greater extent, and better utilising the low vacuum possible with good air pumps and condensing arrangements.

CHAPTER XV.

REGULATING AND EXPANSION VALVES AND GEAR.

Regulating valve.—The steam, after leaving the separator, when this is fitted, arrives at the regulating valve for the engines. The object of this valve is to regulate the supply of steam to the engines, so that the speed may be varied as required.

The regulating valve, as originally fitted, consisted simply of a flat plate or disc in the pipe, having a central spindle passing through to the outside of the pipe, by means of which it could be turned so as to either close or open the passage as required. This is called a 'throttle valve,' but it is now seldom fitted, as it is difficult to keep even approximately tight with high-pressure steam, and it does not admit of sufficiently exact regulation of the speed of the engines.

Flat valves, called 'gridiron valves,' were next used. They consisted of a number of bars with open spaces between them, sliding on a corresponding seating.

As pressures increased, however, their friction became too great, and they were superseded by the 'double beat' or equilibrium valve, now used for regulating purposes.

Equilibrium or double-beat valve.—A sketch of the double-beat regulating valve is shown in Fig. 126. It consists of two valves on the same spindle; the steam pressure acts on the top of one valve and on the bottom of the other, so that the valve is nearly in equilibrium, and little force is required to move it from its seat. The larger diameter of the lower valve in the arrangement shown must obviously be somewhat less than the smaller diameter of the upper valve, to enable the valve to be put in its place. The amount of opening for a certain height of lift is practically double that for the same lift of an ordinary single conical valve of the same diameter.

Manœuvring valve.—Owing to the frequent small changes of speed of engines required in war vessels, when steaming in the company of other vessels, due to the necessity of 'keeping station,' it has been found necessary in these ships to fit a special small valve. This valve is shown at A, in Fig. 126, and it admits steam from one side to the other of the regulating valve. It is found that the latter is so large that a small change in its opening makes a considerable alteration in the speed of the engine when steaming slowly, and small changes of speed are very difficult to obtain by its means.

By means of the small manœuvring valve, however, these necessarily small changes in the speed of the engine to meet the requirements of station keeping are secured. The valve fitted is generally an ordinary screw-down valve.

These regulating and manœuvring valves are worked by screw gearing, so that the speed of the engine may be adjusted with more exactness than is possible with an arrangement of levers. There is also much less back-lash with such gearing than with levers. The wheels leading to the gearing of the regulating and manœuvring valves are placed in convenient situations close to the starting position.

Expansion valves.—Although of little or no importance now for marine purposes, the arrangements fitted in most simple engines, and in some of the early compound engines, to secure economy by using the steam more expansively than could be advantageously effected by the ordinary slide-valves, are worthy of notice. To secure high rates of expansion, separate valves, usually called 'expansion valves,' were fitted for the purpose of cutting off the admission of the steam at a sufficiently early part of the stroke. The sole office of these valves was to

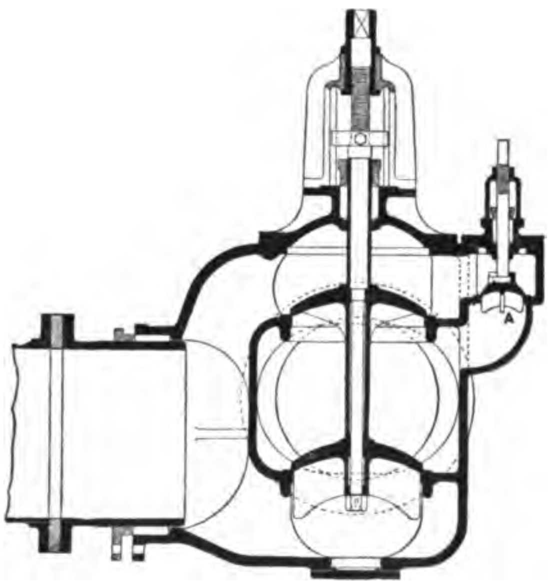


FIG. 126.

stop the admission of steam at the required point at each stroke, and they had nothing further to do with the distribution of the steam, the remaining operations being effected by the slide-valves.

Some early compound engines also had separate expansion valves fitted to the low-pressure cylinder in order to regulate the distribution of work between the two cylinders.

These valves were usually of the gridiron type, a portion of one such valve being shown in Fig. 127. The steam enters the expansion valve casing, and when the valve is in such a position that the ports are open, the steam passes into the slide-valve casing underneath. When the valve is in its middle position all the ports are wide open. A great number of ports are desirable, so that the amount of opening

may be considerably affected by a small motion of the valve, and the cut-off, therefore, sharper and more effective.

Expansion gear.—Expansion valves are worked by eccentrics on the crank-shaft, and the extent of the travel of the valve is regulated by a link, fitted with a sliding block attached to the expansion valve rod. The gear should be so arranged that when the engines are not being worked expansively the motion of the valve should, if possible, be reduced to zero, the valve remaining in its central position, with all the ports wide open, or the motion reduced to such an extent that it has no appreciable effect on the passage of the steam, so that the handling of the engines, going astern, &c., is not affected. By altering the position of the block in the link so as to increase the travel of the valve the point of cut-off is made earlier. The greater the travel the higher will be the rate of expansion.

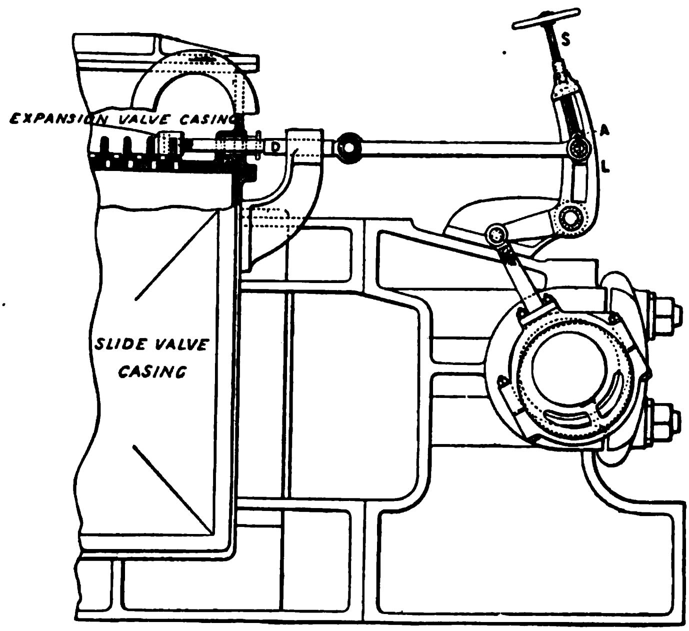


Fig. 127.

Fig. 127 shows the arrangement of expansion gear for a valve of this description as fitted to an old horizontal engine, and the method of its action can be easily seen from the diagram. By altering the position of the block A on the vibrating lever L by means of the screw S, the travel of the expansion valve is altered, and the point of cut-off of the steam regulated as may be required.

Increase of clearance volume.—With an expansion valve such as described above, it will be seen that the clearance between the expansion valve and the main slide-valve was considerable, and when the clearance between the main slide-valve and piston was also added, the total clearance space to be filled with fresh steam at each stroke was

so great as to neutralise much of the advantage due to the use of an expansion valve. The actual amount of expansion was consequently much less than that indicated by the fraction of the cut-off.

Expansion valve on back of main slide.—To overcome this defect the gridiron valve was sometimes fitted to work on ports formed in the back of the main slide-valve, as shown in Fig. 128, the expansion valve being worked as before by a separate eccentric, and the cut-off regulated by the amount of travel of the expansion valve.

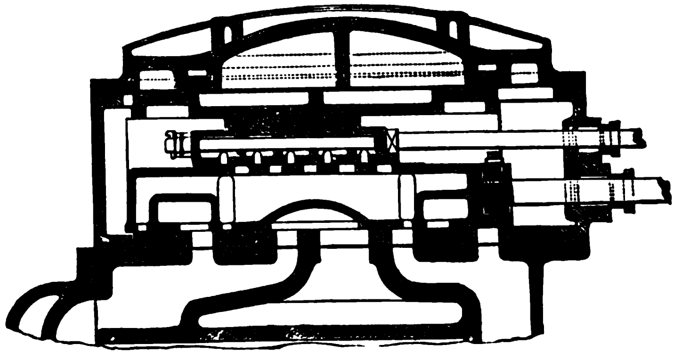


FIG. 128.

Another plan of fitting an expansion valve on the back of the main slide-valve is shown in Fig. 129. In this case the expansion valve was constructed of two separate but similar parts, connected together by a right- and left-handed screw, by means of which their distance from each other might be varied. In this arrangement the travel of the valve was constant, and the point of cut-off was regulated by the distance between the two plates that formed the valve, which could be varied as required by moving the wheel in connection.

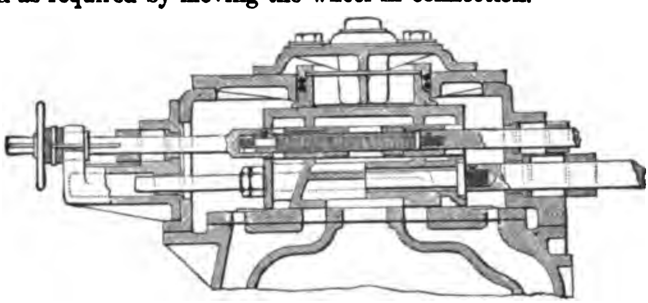


FIG. 129.

So far as the distribution of the steam is concerned these valves, which worked directly on the backs of the main slides, had a great advantage over the expansion valves working in separate casings, as the clearance space that had to be filled with steam at each stroke was much diminished. It was, however, often found difficult to properly lubricate the working surfaces, especially in large engines, and the valves wore away rapidly, causing excessive stresses on the gear from

the great friction, and for this reason the most commonly employed expansion valve was that working in a separate casing, as in Fig. 127, which gave less trouble.

Reasons for abandonment of separate expansion valves.—The separate expansion valves when fitted were always troublesome to maintain in proper order. As steam pressures increased and larger ratios of cylinder were fitted in compound engines, the necessity for such valves passed away. It was found that most of the expansion capable of being usefully carried out could be obtained with the cut-off of the ordinary slide-valve in association with the stage expansion system, so that for a steam pressure of 90 lbs. the expansion valves were abandoned. The usual ratio of H.P. to L.P. cylinder-volumes with this pressure was 1 to 4, so that a fair amount of expansion was

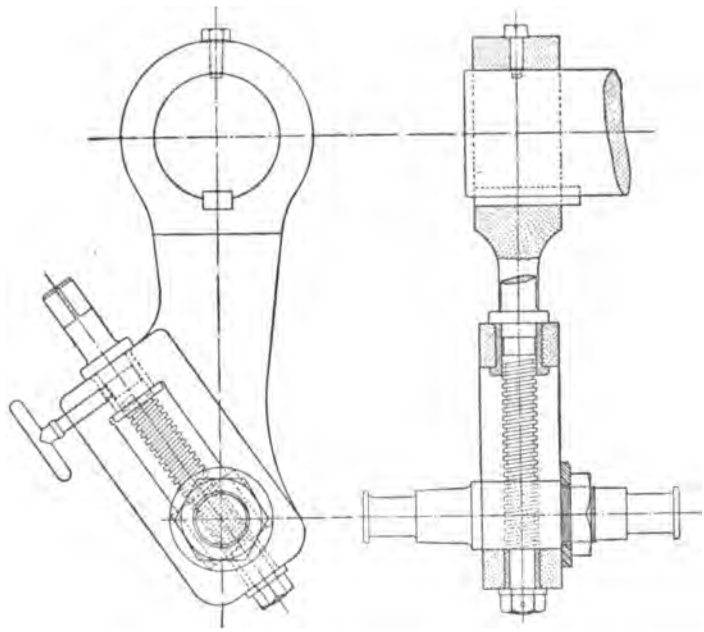


Fig. 130.

obtainable within the limits of cut-off which could be given efficiently with the ordinary slide-valve.

With the modern triple and quadruple expansion engines there is still less necessity for separate expansion valves, and they are not fitted, as all the expansion required is obtainable by means of the link motion.

Independent linking-up gear.—All modern engines in the Royal Navy which are intended to work with considerable variations in the total power are provided with means for independently linking up the slide gears of the various cylinders. This gear enables the cut-off in any cylinder to be made earlier than that corresponding to the position of the main reversing gear. This latter gear, when moved by the starting-

engine or wheel, alters the cut-off in the various cylinders to the same extent, but the independent linking-up gear enables various alterations of cut-off to be made in any cylinder, so that the total power at any speed may be more equally divided between the various cylinders, and any other desirable adjustments made. The effect of these changes on the distribution of power in the various cylinders is explained in Chapter XXVI.

To enable this to be effected, the reversing arms attached to the weigh-shaft are fitted with slots and sliding blocks to which the suspension rods leading to the links are attached. By moving these blocks the links are altered in position. As the alteration of the blocks takes some time, the angle of the slot is so arranged that in the astern position the slot is approximately perpendicular to the suspension rods, so that the position of the block in the slot does not affect the link in the astern position. This being so, the engines may be reversed without making any alterations in the independent linking-up gear. A sketch of this fitting is shown in Fig. 130. A nut and washer are provided on one side of the block pin, which when tightened up secure it in position. This nut is slacked back before any alterations in the position of the block are made.

CHAPTER XVI.

SLIDE-VALVES AND FITTINGS.

Slide-jacket.—The steam, after passing the regulating valve, enters the slide-jacket or casing, which is simply a rectangular or cylindrical box bolted to the cylinder, in which the slide-valve works. This slide casing is either cast in one with the cylinder, or bolted to it.

Slide-valve.—The distribution of steam in each of the steam cylinders of an engine, involving the processes of admission, expansion, and finally exhaust into the receiver pipes of the succeeding cylinder or the condenser as the case may be, is now effected by the agency of a single valve, called the 'slide valve.' The slide-valve is one of the most important parts of the engine, and on the skill and care exercised in its design and fitting, the satisfactory working of the machinery will greatly depend.

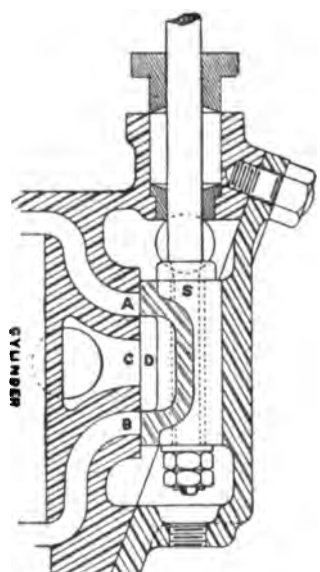


FIG. 131.

A section of an ordinary single-ported slide-valve of a small engine is shown in Fig. 131. It is kept in position on its cylinder face by the cover shown. The inclination shown for the joint between cover and cylinder is desirable, as this renders the cylinder face more readily accessible for scraping or filing up to a true plane surface.

In the cylinder face there are three passages, called 'ports,' marked respectively A, B, and C. A and B are the *steam* passages or ports, one leading to each end of the cylinder, and C is the *exhaust* port leading steam from the cylinder. S is the slide-valve, which is rectangular in plan, and the hollow space, D, in its centre is called the *exhaust cavity* of the valve. The valve is shown in its central position, and it will be seen that it not only closes the steam ports, but overlaps the edges for some distance on the outside or steam side. The object of this will be explained later.

The slide-valve has a flat face, and it works steamtight on the corresponding flat face of the cylinder. The casing around it is supplied with steam, while the exhaust port is connected either to the condenser, the reservoir, or the atmosphere, depending on the type of engine.

Action of the slide-valve.—We will examine first the motion of such a valve and the distribution of steam in the cylinder during one revolution of the engine, noting the movements of the piston at the same time.

We will assume the piston to be just commencing its stroke, it being always arranged that the slide-valve shall then have uncovered the steam port by a certain distance called the '*lead*,' this condition being represented in Fig. 132. Steam enters through the port A and pushes the piston in the direction of the arrow, the valve moving also in the same direction. The other port, B, is open on the inside edge, and exhausts the steam on the side E of the piston to the exhaust pipe shown in the exhaust cavity C. When the piston and valve have travelled a certain distance in the direction indicated, the valve reaches the end of its travel and is for an instant at rest, the piston, however, continuing to move in the same direction as shown in Fig. 133. The exhaust port, B, is now wide open, but the valve and

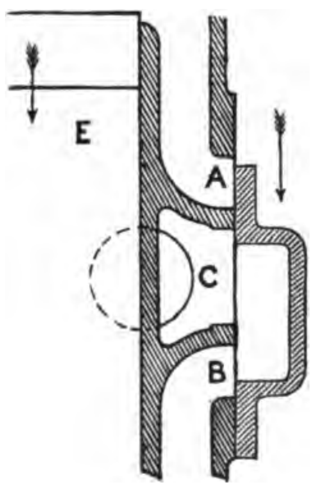


FIG. 132.

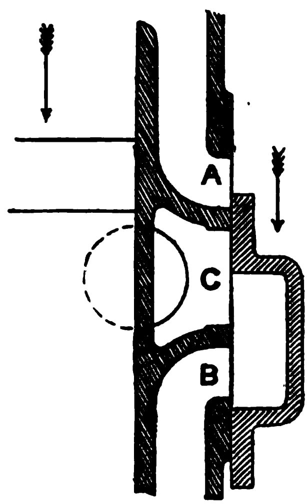


FIG. 133.

ports are generally arranged so that the ports are never wide open to the steam, so that A is not wide open. The reason for this will be explained later.

The valve now commences its return stroke, the piston and valve travelling in opposite directions, the next important phase being shown in Fig. 134, when the admission of steam to the cylinder is stopped by the steam edge of the valve closing the port A. This is called the *instant of 'cut-off,'* and the remainder of the piston's motion in the direction of the arrow is caused by the expansion of the steam previously admitted. It will be noticed that the port B is still open to exhaust. The piston and valve now proceed still further in opposite directions until the piston has travelled nearly the whole of its stroke and the valve reaches the middle of its travel, as

in Fig. 135. In this position the two inner or exhaust edges coincide with the inner edges of the port.

Two important operations now occur. On the side E of the piston the steam or vapour which had previously been passing out through the port B into the exhaust pipe is now confined by the closing of the port, and as the piston proceeds further in the same direction the steam still remaining in the end E of the cylinder is compressed, and its pressure will gradually increase as the piston gets nearer the end

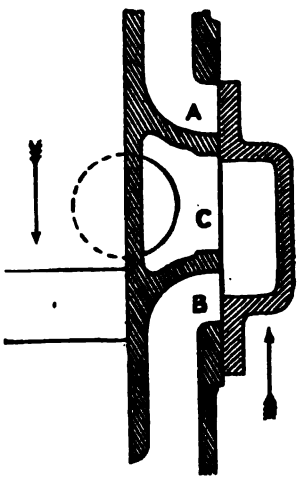


FIG. 134.

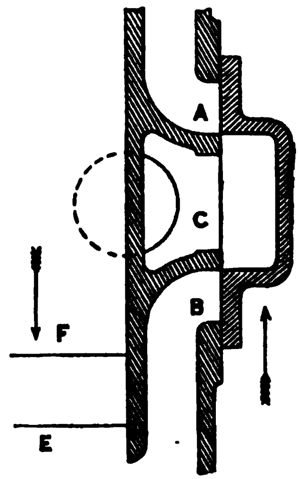


FIG. 135.

of its stroke. This is called the *instant of 'compression.'* Its effect is to provide an elastic cushion of steam to absorb the momentum of the piston and parts attached, and bring them to rest gently before the opposite stroke is commenced, thus avoiding shocks, and assisting the entering steam to start the piston on its return stroke. It has also an important effect as regards efficiency of the steam, as by its means the clearance spaces are filled with compressed steam, and a smaller quantity of steam from the steam pipe is thus required for each stroke. See Chapter XXVI.

On the other side, F, of the piston another important operation also occurs, for the valve is still travelling in the direction of the arrow, and the inner edge of the valve now commences to open the port A to the exhaust pipe, and the steam which has previously been driving the piston forward by its expansive force now rushes off to the condenser, and the pressure on the side F is suddenly reduced. This is called the *instant of 'release.'* It will be noted that with the valve as shown, having both its exhaust edges exactly corresponding to the exhaust edges of the cylinder ports at the same time, the operations of 'exhaust' on one side of the piston and 'compression' on the other occur at the same instant. If, as is often the case, these edges do not correspond exactly, the exhaust and compression will occur at different instants.

As the piston travels still further towards the end of its stroke, the valve proceeds in the direction of the arrow, and rapidly uncovers the port A to the exhaust pipe, and also the compression on the side E proceeds till just before the piston reaches the end of its stroke, when the steam edge of the valve reaches the edge of the port B and commences to admit steam. This is termed the *instant of 'admission'* (Fig. 136). The pressure on the side E then rises to the full steam pressure, and the small remaining part of the piston's stroke is completed against this steam pressure, which continues the action of the compressed steam in bringing the piston gradually to rest prior to the commencement of the return stroke.

When the stroke of the piston is completed, as in Fig. 137, the valve is again open to steam by an amount equal to the 'lead,' and the operations described above are repeated on the opposite side of the piston, while the latter performs the return stroke, until the piston and valve are again in the same position as in Fig. 132. The steam is thus

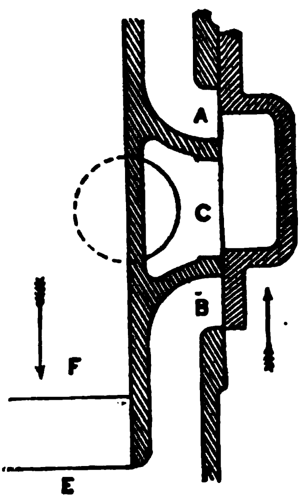


FIG. 136.

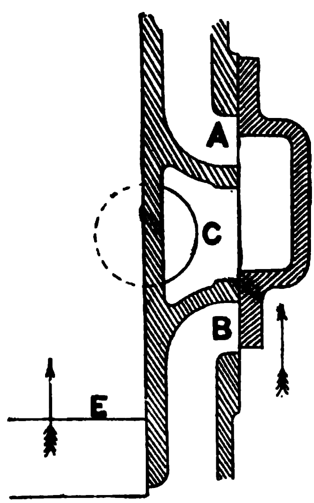


FIG. 137.

admitted to, and exhausted from, the opposite ends of the cylinder, and a motion of the piston to and fro in the cylinder caused, and this reciprocating motion of the piston is communicated to the crank-shaft of the engine by the mechanism described in Chapter XXI, the shaft being thus continuously rotated so long as steam is supplied to the cylinder. The effect of arranging the slide-valve with 'lead' is to considerably increase the opening to steam when the piston is commencing its stroke, so as to assist in avoiding any considerable fall of pressure due to contraction of the steam inlet at this period. It also, as explained above, allows steam to be admitted just prior to the completion of the stroke, and thus helps in bringing the piston gradually to rest and avoiding shocks at the end of the stroke.

It should be clearly noticed that the points of admission and cut-off are determined by the steam edge of the valve, and those of release and compression, by the exhaust edge.

In considering the motion of the slide-valve the student will find it a very instructive exercise to draw to scale a section of the cylinder ports as shown in the diagrams, and make a cardboard model of the section of the slide-valve, so that it may be worked over the cylinder ports as desired.

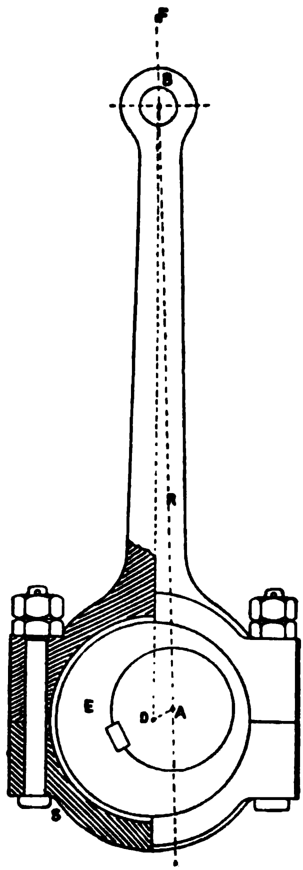


FIG. 138.

Motion of the slide-valve, eccentric, and eccentric rod.—The motion of the slide-valve to and fro, causing the reciprocating motion of the piston, is generally produced by means of an eccentric and rod, sketches of a small example of which are given in Fig. 138. A circular cast-iron sheave, E, has bored in it, eccentrically with its own circumference, a hole of the same diameter as the crank-shaft. This eccentric sheave is keyed firmly on the shaft, so as to revolve with it. The centre of the eccentric is indicated at D, while A is the centre of the shaft, which remains fixed, and about which centre the shaft, carrying with it the eccentric, rotates. On the circumference of the eccentric there works a ring, S, called the eccentric strap, to which the eccentric rod, R, is attached. The end, B, of the eccentric rod is connected by a joint to the slide-valve rod. In large marine engines this connection is not made direct, but through the agency of a 'link,' as explained in Chapter XVII.

When the shaft revolves, carrying the sheave with it, as the end B of the eccentric rod is prevented from moving except along the line A F, the sheave must slide in the strap and sway the latter to and fro, thus producing a reciprocating motion in the end B, of the eccentric rod, and consequently in the slide-valve itself, to which it is connected.

The extent of the travel of the end B of the rod along the line A F is evidently equal to twice A D; i.e. twice the distance between the centres of the crank-shaft and eccentric sheave. This distance A D is called the '*eccentric radius*' or '*eccentric arm*,' or '*throw of eccentric*.'

Eccentric and rod equivalent to crank and connecting rod.—This motion is evidently the same as that outlined in Fig. 139, where the eccentric sheave and shaft are replaced by a solid plate revolving about the point C. The eccentric rod E D is evidently always normal to

the circumference of the revolving plate, and therefore *always points to its centre*, P. As C P is a constant distance, the motion is thus equivalent to that which would ensue were the point E connected to C by means of a revolving crank, C P, and a connecting rod of length E P. The eccentric and rod are therefore clearly equivalent in action to that of a small crank and a connecting rod, and are adopted in cases where, from the smallness of the travel, it is inexpedient to obtain the motion by the direct intervention of a crank.

It is important, in examining the action of the slide-valve, to carefully consider this motion. It will be seen that the longer the connecting rod, P E, the more nearly does its direction become parallel to the line of motion, C E, of the end of the connecting rod. When this length becomes infinitely great the direction of P D becomes parallel to C E. The motion is then equivalent to that obtained by the eccentric acting against a flat bar, A B (Fig. 140), at the end of a sliding rod, S, which

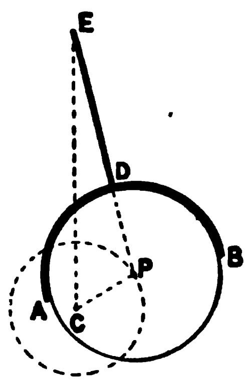


FIG. 139.

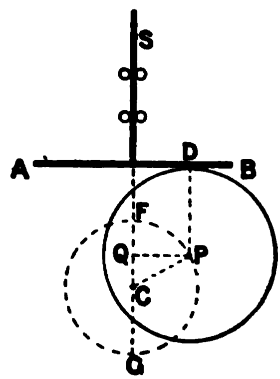


FIG. 140.

is kept moving in a vertical straight line, S C, by suitable guides. The point of contact, D, of the bar with the cam, will always be vertically above the centre, P, and the vertical motion of the bar is then exactly the same as that of the point P, and therefore also of the point Q, the foot of the perpendicular drawn from P on the line C E.

Harmonic motion.—As the length of the eccentric rod in well-designed gears is large compared with that of the eccentric radius, its influence in causing a deviation from the motion illustrated in Fig. 140 is not very great, and for most practical purposes the motion of an ordinary slide-valve may be assumed to be as shown in the last-mentioned Figure, and its geometrical representation and examination is thereby much facilitated. If the point P is made to revolve uniformly about the centre C, the motion of the point Q along the line G F is described geometrically as an exact 'harmonic' motion. It will be seen that its velocity reaches a maximum when passing the centre, C, while near the ends of the stroke the velocity gradually lessens, till it becomes zero when the point P arrives at F and G the ends of the stroke. Q is then for an instant at rest while its motion is being reversed.

The slide-valve is therefore at the centre of its stroke when the eccentric arm makes an angle of 90° with the line of motion, and further, as C Q = C P cosine P C Q, the distance of the slide-valve from its central position is equal to the eccentric radius multiplied by the cosine of the angle this radius makes with the line of motion of the end of the rod, or line of dead centres.

Geometrical representation.—This motion is capable of simple geometrical representation as follows. We have seen that when the eccentric radius is at C P, the slide-valve is distant C Q from the centre of its stroke. Suppose we mark off along C P a distance C Q′ = C Q. If we do this for all positions of C P, and draw a curve through all the points such as Q′, we obtain two circles with diameters C F and C G (see Fig. 141, in which for clearness the circle F G has been enlarged). This is easily proved, for if F Q′ be joined the two triangles C Q P and C Q′ F are equal in all respects, therefore the angle F Q′ C, which is equal to the angle P Q C, must be a right angle; hence the point Q′ must lie on a circle with C F as diameter. These circles are such, therefore, that if any position of *eccentric radius* such as C P be drawn, the part C Q′ intercepted by the circle gives us the distance the slide-valve has moved from its central position when the *eccentric radius* is in the position C P. The diameter of each of these circles is the half travel of the slide-valve.

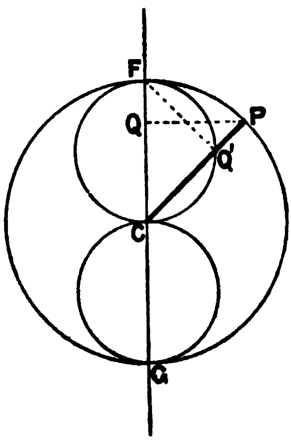


FIG. 141.

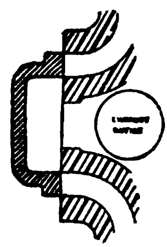


FIG. 142.

Slide-valve without lap or lead.—The most simple form of the slide-valve is a single-ported valve, without either lap or lead, as shown in Fig. 142. It is clear that any slide-valve must be long enough to cover both ports on the steam side at the same time, or otherwise the steam would pass to both sides of the piston at once, and no motion would ensue. In the present example the valve is just of sufficient length to exactly cover both the ports.

In the figure the valve is shown in the centre of its stroke, just closing both steam ports, while at this instant the eccentric radius must be perpendicular to the line of motion.[1] The piston is clearly at the commencement of its stroke, and the crank on its dead point. We see, therefore, that in such a case the eccentric must be fixed on the shaft in such a position as to make an angle of 90° with the crank. As the

[1] See top of this page.

crank and eccentric revolve, the valve begins to admit steam to one side of the piston, and to place the other side in connection with the condenser through the exhaust passages, so that the steam behind the piston may escape. The amount of opening continues to increase, till the piston arrives at half-stroke, when the steam port is wide open. After this it begins to close, but does not shut completely till the piston arrives at the end of its stroke.

With this arrangement, the valve begins to open the ports, both to the steam pipe and condenser, at the beginning of the stroke; the ports continue open to a greater or less extent during the whole period of the stroke, and there is no expansion.

Reversibility with single fixed eccentric.—This simple form of slide-valve is one very commonly used for small auxiliary engines, for by a simple arrangement it can be made reversible, although it has only *one fixed eccentric*. To enable this to be seen Fig. 143 has been drawn, showing the valve after it has performed a part of its stroke downwards from the middle position, while below it is shown the corresponding position of eccentric radius and crank. If, as is commonly the case, the space A is in connection with the steam pipe, and B in connection with the exhaust, the steam would in this position enter the top of the cylinder, causing the piston to descend, and the motion of the crank to be as indicated by the arrow in full lines. Conversely, if the space B be supplied with steam, while A is connected with the exhaust pipe, steam will enter the bottom of the cylinder through the inner edge of the lower port, and the piston will be forced upwards, and motion will ensue in the opposite direction, viz., that indicated by the dotted arrow. The last-mentioned arrangement of steam and exhaust supply to the valve is sometimes arranged for permanently; but in the case more immediately under consideration, viz., that in which reversibility is required by means of a simple arrangement, a device is adopted by which the steam and exhaust pipes can be placed in connection with A and B respectively, or interchanged as desired, and by this means the engine is made to run in either direction.

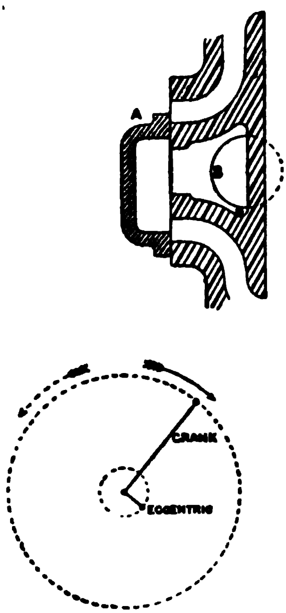


Fig. 143.

The apparatus by means of which this interchange of steam and exhaust is effected is described elsewhere.[1] It will be seen, therefore, that with a valve of this description the eccentric radius is fixed at 90° in advance of the engine crank when the steam is supplied to the outside edges of the valve, while it is 90° behind the engine crank when the steam is supplied to the inside edges of the valve. All reversible engines with a single eccentric to each cylinder are fitted

[1] See Chapter XVII.

with valves of this character, such as steering engines, capstan engines, starting engines, turning engines, &c. These valves are generally of the piston type.

It may be mentioned in passing that as the slide-valves are practically always open at each end of the cylinder, either to steam or exhaust, drain-cocks are often omitted from these engines, the accumulated water being forced by the piston into the exhaust pipe on starting the engine.

Having described the elementary slide-valve, we proceed now to consider the most usual type of slide-valve in more detail, first explaining a few definitions.

Lead.—In ordinary engines, to facilitate their working, the slide-valves are arranged so that they may open both to steam and exhaust shortly before the end of the stroke. This is done by advancing the position of the eccentric arm with respect to the crank, so that all the motions of the valve may be earlier.

If this be done with the valve shown in Figs. 142 and 143, it will be seen that although the valve opens just before the end of the stroke, yet it does not produce expansive working of the steam, as the ports are still open either to steam or exhaust for the whole duration of a stroke.

The lead of the valve is defined as the length of opening of port to steam at the *beginning of the stroke of the piston.*

Lap.—It is found in practice that, in order to produce smooth and economical working, it is necessary to provide means for cutting off the admission of the steam before the end of the stroke. This is accomplished by giving the valve *lap* or *cover*, or, in other words, by lengthening the valve so as to make it more than exactly cover the ports when in its middle position, as shown in Fig. 144, in which one end only of the valve is shown. P represents the steam port of the cylinder, and S the section of the slide-valve.

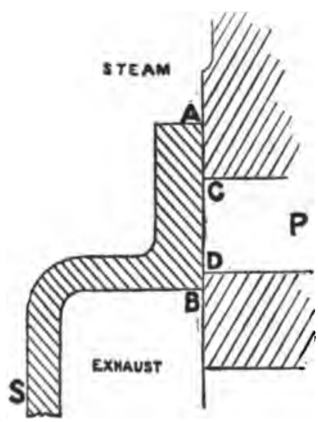


FIG. 144.

The side of the steam port of the cylinder at which the steam enters is called the induction or steam side, and the side at which it begins to exhaust the eduction or exhaust side. Similar names are given to the corresponding edges of the slide-valve itself. For example, in Fig. 144, assuming steam to be supplied to the outside of the valve, C is the induction or steam side, and D the eduction or exhaust side of the cylinder port, and A the induction and B the eduction edge of the slide-valve.

The lap or cover of a slide-valve is the extent to which the edge of the valve overlaps the adjoining edge of the cylinder port *when the valve is in the middle of its stroke.* For example, A C is the lap on the steam or induction side and B D the lap on the exhaust or eduction side of the slide-valve. The lap on the steam side is generally called the outside lap, and that on the exhaust side the inside lap of the valve.

The lap on the exhaust side when fitted is always small, and is often absent altogether, as it is important that the communication with the condenser should be as free as possible. In many cases, especially in fast-running engines, *negative lap* on the exhaust side is fitted to increase the length of time the cylinder is open to the condenser during each stroke. In Fig. 145 B D is the amount of negative inside lap.

Giving outside lap to the valve will, in the first place, necessitate an increased travel, to obtain the same amount of opening for the steam; for it is clear that the valve must travel through a distance equal to its outside lap before it begins to open the steam port at all. Secondly, in order to give the necessary lead at the beginning of the stroke, the eccentric will have to be still further advanced with respect to the crank than in the case of a valve without lap, and consequently all the movements of the valve will be earlier than before. The effect of lap on the exhaust side or inside lap will be to close the communication with the condenser earlier than would otherwise be the case, so that a larger quantity of steam would be confined in the cylinder, and compressed behind the piston, until the end of the stroke.

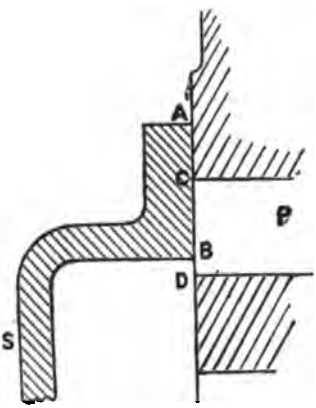


FIG. 145.

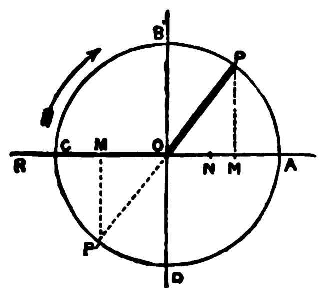


FIG. 146.

Angular advance of the eccentric.—In a slide-valve without lap or lead we saw that the eccentric arm was at right angles with the crank [1]; but in the ordinary slide-valve with lap and lead, the action of which was described on page 173, it is necessary, in order to give the valve the required lead at the beginning of the stroke, to considerably advance the eccentric beyond this position, and the amount it requires to be moved is termed the '*angular advance of the eccentric.*'

In ordinary engines, therefore, in which the slide-valves work parallel to the pistons, the angular advance is the angle between the eccentric radius and a perpendicular to the crank arm. To meet the exceptional cases in which the slide-valves do not work parallel to the pistons, the term may be defined as follows:—When the crank is on the dead centre, the angle at which the eccentric radius stands in advance of the position that will bring the valve to its mid stroke is called the 'angular

[1] See p. 178.

advance of the eccentric.' These definitions apply to all cases, whether the steam be taken on the inside or outside edges of the valve.

The angular advance of the eccentric may be obtained, approximately, as follows :—Draw a circle, A B C D, Fig. 146, with the length of the eccentric arm as radius. Let A R be the line of motion of the piston, and O R the position of the crank when on the dead centre. If steam is taken on the *outside* edges of the valve, set off *away* from the crank, O M = the outside lap, plus the lead. Draw M P vertically, cutting the circle in P. Then O P will be the approximate position of the eccentric radius when the engine crank is on the dead centre, and the angle B O P will be the angular advance of the eccentric. This follows from the fact that when the crank is on the dead centre the valve is distant from its middle position by an amount equal to the lap plus lead.

If, however, steam is taken on the inside edges of the valve—as is often the case, especially with piston valves—the lap plus lead must be marked off in the opposite direction, viz. *towards* the crank, and O P′ will then be the position of the eccentric, and D O P′ the angle of advance. O D is then the eccentric position if without lap or lead, and, to provide this, the eccentric arm is 'advanced' from O D to O P′.

Opening of the steam port.—The opening of the port to steam, at any instant, is equal to the distance the valve has moved

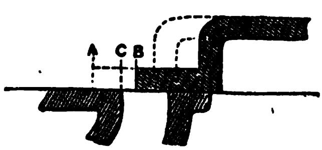


FIG. 147.

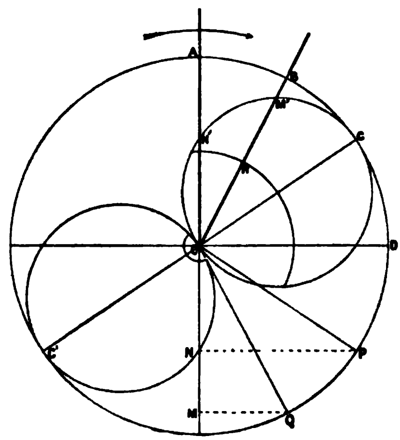


FIG. 148.

from its central position, minus the lap of the valve. Let the dotted lines in Fig. 147 represent the valve in its central position, and the full lines its position at any given instant. Then A B is the distance the valve has travelled from the middle position, and A C is the outside lap.

Therefore Opening of port = B C = A B − A C

= movement of valve from middle position − outside lap. . . (1).

Zeuner's valve diagram.—The simple diagram (Fig. 141) gave for each position of *eccentric radius* the distance of the slide-valve from its middle position, but what is generally required is a diagram giving the position of the slide-valve for each position of the *crank arm* and piston.

Suppose the crank on the dead centre in the direction O A, Fig. 148, and the eccentric radius in the direction and equal to O P, then P O D is the angle of advance, and the valve in this position is distant O N from its central position. Set off O N′ along O A equal to O N. Next, suppose the crank has moved to O B, then the eccentric radius has moved to O Q

such that angle A O B = angle P O Q, and O M is the distance of the valve from middle position. Set off O M′ along O B equal to O M. If this is done for all positions it will be found that all the points such as N′, M′, &c., lie on two circles with diameters O C and O C′, which are each equal to the half travel of the valve, and make an angle with the line of dead centres measured in the direction of motion of the crank equal to 90° minus the angle of advance. We now have a diagram which gives for each position of the *crank*, the distance of the slide-valve from its central position.

If now a circle be drawn with centre O and radius O H equal to the outside lap, the amount of opening of steam port for any position of crank, such as O B, must be given by the part H M′. This follows from equation (1) above, since O M′ is the movement of the valve from middle position. From these circles we are enabled to completely analyse the motion of the slide-valve. The construction is due to Zeuner.

The complete Zeuner's diagram is given in Fig. 149, and this should be carefully studied. To avoid complication it is drawn for one side of the piston only, viz. the top side, and all dimensions such as lap, lead, &c., are also for the top of the valve.[1] The upper circle is termed the 'steam circle' and the lower the 'exhaust circle.'

Now admission and cut-off of the steam occur when the valve is distant from its middle position by an amount equal to the outside lap, i.e. the intersections of the lap circle with the valve circles give the positions of the crank when admission and cut-off occur. Similarly *on the other side of the middle position of the valve*, as release and compression occur when the valve is distant from its mid position by an amount equal to the inside lap, the positions of crank at release and compression are given by drawing an arc of a circle with centre O and radius equal to the inside lap, and the intersection of this arc with the opposite circle gives the positions of the crank at release and compression. O P is the position of the crank at release, and O M the position when compression takes place.

If, as is often the case, the inside lap be negative, the intersection of the inside lap circle with the other circle, O G C, must be taken to obtain the positions at release and compression. The lines O P and O M will then lie on the other side of the perpendicular O N.

For any position of crank O S we see that T S is the amount of opening of valve to steam. The intercepts on the shaded area of the top circle all represent steam openings. Therefore F G is the lead. From the position O B to O C we see that the opening to steam continuously increases, at first quickly, and finally slowly, while from O C to O D the valve is closing at first slowly, and finally rapidly.

Similarly on the exhaust circle the intercepts on the shaded area represent exhaust openings. As, however, the valve is fully open to exhaust when it has travelled a distance from the centre equal to inside lap + the width of the port, we must draw an arc H K on the exhaust circle with this as radius to give the outside limit of the width of opening. Between O H and O K the valve is fully open to exhaust, from O P to O H it gradually opens, and from O K to O M it gradually closes.

The angle C O Y is the angle of advance. A line drawn through

[1] They are often different for the two ends of the valve.

O perpendicular to C C′ gives the positions of crank when the valve is in its middle position.

There are some useful geometrical facts to be observed in this diagram, attention to which will facilitate the solution of problems. Join D B, then this line is a tangent to the cut-off circle. Also draw A L perpendicular to D B, then A L = F G = lead.

Again, if O N be perpendicular to C O C′, a perpendicular from N on

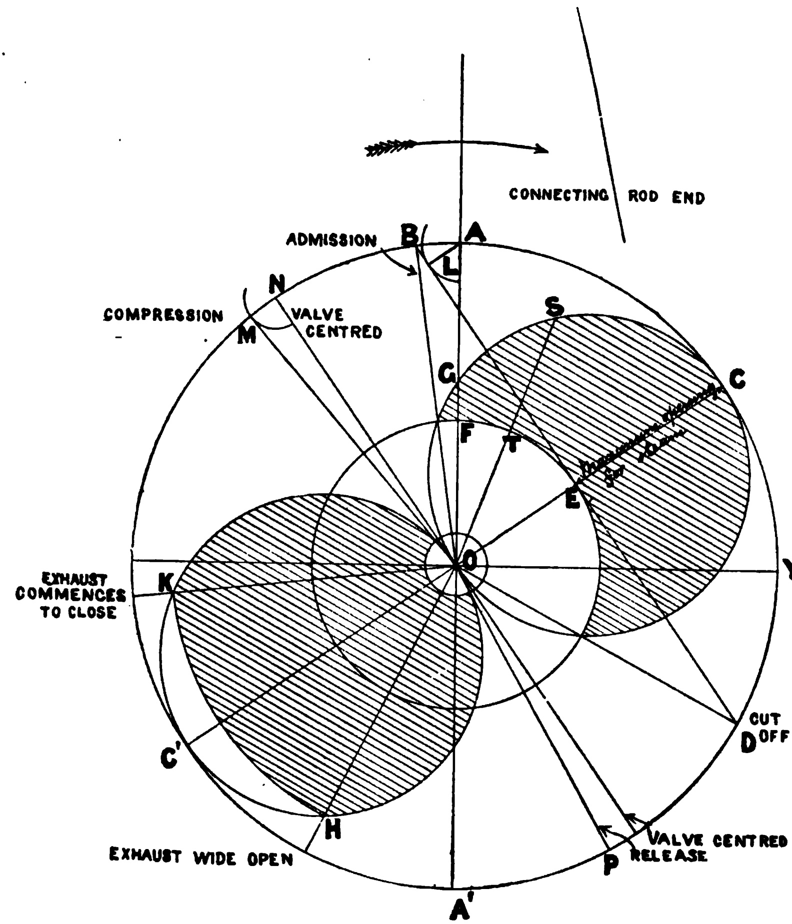


FIG. 149.

O M the compression line, will be equal to the inside lap. These propositions can be easily proved.

The latter fact is generally used for ascertaining more accurately the inside lap from the position of release or compression, or *vice versâ*, as the liability to error is less, it being sometimes difficult to determine the exact point of intersection of the compression line with the exhaust circle.

If now we let A A′ represent on some other convenient scale the stroke of the piston, the mean indicator diagram to be expected can be drawn, neglecting the obliquity of the connecting-rod, by ascertaining by projection on a vertical line the positions of the piston corresponding to points of admission, cut-off, release, and compression.

As regards the diagram for the bottom of the piston, the steam circle for top of piston becomes the exhaust circle for the bottom, and *vice versâ.*

If the inside and outside laps are the same for bottom as for top, the same lap circles would be continued to the opposite circles to give the corresponding points for the bottom of the piston. Often, however, they are different, when, of course, the correct radii must be used.

An elementary knowledge of geometry will, by the application of this diagram, lead to the solution of most of the problems relating to the motion of slide-valves. By varying some of the points the alteration in the others may easily be found, and by assuming certain elements the remainder may be determined.

Position of piston.—As we have stated, the influence of the length of the eccentric rod may be neglected in considering the motion of the slide-valve, unless this length be much shorter than is usual in practice as compared with the eccentric radius. The connecting-rod is, however, always much shorter relatively to the crank-arm, and it must always be taken account of in considering the distribution of steam by the slide-valve. The motion of the piston is evidently the same as that of the crosshead at the end of the connecting-rod. Suppose X′ P X, Fig. 150, to be the path of the crank-pin, and let O P and O Q be two positions of crank-pin at equal angles with the line of dead centres. When the crank is at P and Q respectively, the position of the crosshead is found by drawing from these points as centres, arcs with radius equal to the connecting-rod, cutting the line of centres. Suppose A and B be the positions thus found, and let Y and Y′ be the ends of the stroke of the crosshead. With centres A and B draw arcs P S and Q T; then X S and X′ T will be the distances of the piston from the ends of its stroke. This is easily seen since S A = P A = X Y ∴ S X = A Y.

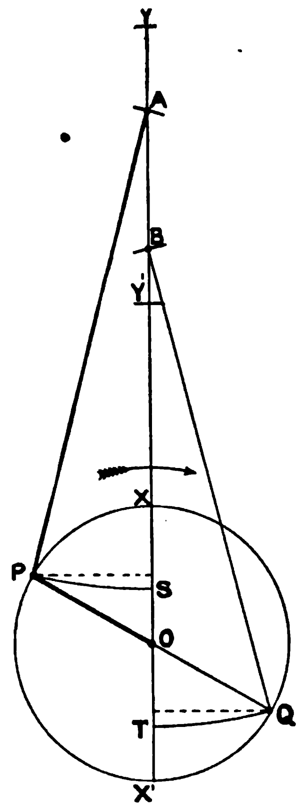


FIG. 150.

Suppose, for example, Q and P are the positions of the crank at cut-off on the down stroke and up stroke respectively, then it will be seen that the piston on the down stroke has at the point of cut-off travelled a distance X T, which is greater than the distance X′ S which has been travelled on the up stroke. For the same position of crank

at cut-off, therefore, more steam will be admitted to the cylinder on the down stroke than on the up stroke. The piston, except when on a dead centre, is, in fact, owing to the influence of the connecting-rod, always nearer the crank-shaft end of the stroke than it would be if the connecting-rod were infinitely long. In the latter case, the distances travelled during the up stroke and down stroke would be equal, and would be represented by the feet of the perpendiculars from P and Q respectively, as shown by the dotted lines.

Having obtained in Fig. 149 the positions of the crank for the various operations of the slide-valve, the positions of the piston can be obtained by drawing arcs, such as P S and Q T of Fig. 150, and this is necessary before the fraction of stroke performed at cut-off, &c. can be given. The geometrical method described below may also be used.

The following table shows the relative fractions of the stroke traversed for the particular, but very common, case in marine engines, of the connecting-rod being equal in length to four times that of the crank :—

Down Stroke or Outward Stroke		Up Stroke or Inward Stroke	
Angle of Crank	Distance travelled	Distance travelled	Angle of Crank
0	0	1·0	180
10	·009	·991	170
20	·037	·963	160
30	·082	·918	150
40	·143	·857	140
50	·215	·785	130
60	·297	·703	120
70	·384	·616	110
80	·474	·526	100
83	·500	·500	97
90	·562	·438	90
100	·647	·353	80
110	·726	·274	70
120	·797	·203	60
130	·868	·132	50
140	·914	·086	40
150	·948	·052	30
160	·977	·023	20
170	·994	·006	10
180	1·000	0	0

It will be noticed that to bring the piston to its mid stroke the crank has to travel through only 83° on the down stroke, while on the up stroke this is not effected till the crank has traversed 97°.

Geometrical representation.—This is well represented graphically by another diagram [1] :—

Suppose in Fig. 151 that O O_1 = the length of crank-arm,

O_1 A = the length of connecting-rod ;

with centre O and radius O A describe a circle, and with centre O_1 and radius O_1 A describe another circle. If O R be any position of the crank, S R the intercept between these circles will be the distance travelled by the piston. By joining O_1 S it is easily seen that as O_1 S =

[1] Due to Müller.

connecting-rod length, the triangle $o\,o_1\,s$ is an exact reproduction of the relative positions of the crank, connecting-rod, and line of motion of piston when the crank has turned through the angle $o_1\,o\,s$, so that $o\,s$ is the distance of the crosshead from the centre of the shaft.

The difference $o\,r - o\,s = s\,r$ must therefore be the distance the piston has travelled from the top of the stroke.

By drawing another circle with centre o and radius $= o\,b$, we get $t\,r = b\,c =$ full stroke, so that $t\,s$ will be the distance of the piston from the other end of the stroke, and if $o\,r$ be the position of the crank on the up stroke, $t\,s$ will be the distance it has travelled from the bottom position.

For any position of the crank, therefore, we have the exact position of the piston given by the point s, the outside intercept $s\,r$ giving the distance from the top of the stroke, and the inside intercept the distance from the bottom of the stroke.

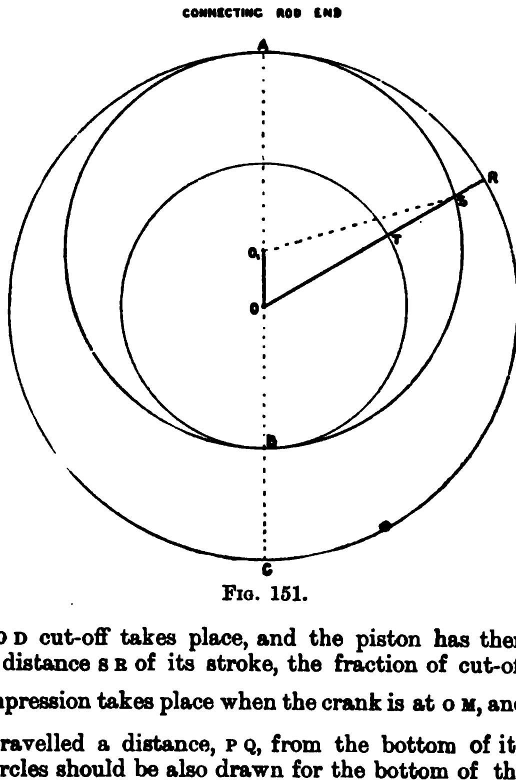


FIG. 151.

The diagram may conveniently be superimposed on Zeuner's valve diagram to such a scale as to be clear of the latter, as shown in Fig. 152, and by this diagram the exact position of the piston and valve for any position of crank can be accurately studied.

The lap circles are drawn for the *top* of the piston, so that, for example, when the crank has turned to $o\,d$ cut-off takes place, and the piston has then travelled through the distance $s\,r$ of its stroke, the fraction of cut-off being $\frac{s\,r}{t\,r}$. Again compression takes place when the crank is at $o\,m$, and the piston has then travelled a distance, $p\,q$, from the bottom of its stroke. Similar lap circles should be also drawn for the bottom of the piston, but this is omitted on the diagram for clearness.

It will be seen, therefore, that if the outside lap be the same at each end, the position of piston at cut-off will be later, and therefore the power exerted will be greater in the stroke towards the crank than stroke from the crank, and this is aggravated in vertical engines by the weight of the reciprocating parts, which act in the same direction.

For this reason, therefore, in large vertical engines the outside lap on the top side is made greater than that on the bottom in order to rectify the inequality. This is often done by simply shifting the

valve a small distance upwards along its rod, which also increases the lead and the inside lap at the bottom end, which gives more cushioning at this end, thus counteracting the descending weight of the piston, rods, &c.

Double- and treble-ported slide-valves.—When the cylinders are large, it is found that the single-ported slide-valve shown in Fig. 131 would necessitate a very great travel, which would be inconvenient in practice, and cause much extra work. To meet this difficulty, double-ported valves, and sometimes treble-ported ones, have been introduced. Sketches of a double-ported valve are given in Figs. 153 and 154, Fig. 154 being a cross section through the dotted line at B E. These valves are fitted to all large marine engines. Their action in the distribution of the steam is the same as that of the single-ported valves, but the details of their construction are different. The steam passage at each end of the cylinder, instead of terminating in a single port in the cylinder face, is divided into two parts, each being one-half the width necessary for a single port; so that the travel of the slide-valve, to admit a given quantity of steam, is only one-half of that required for the ordinary valve.

In the single-ported slide-valve the steam is only admitted at the ends, and enters the cylinder when the valve has moved a sufficient distance to allow the steam to pass from the outside of the valve to the steam port of the cylinder. The outside edges of the double-ported valve act in a similar manner, but, in addition, there is what is practically an inner valve, to which steam is admitted through the passages A A, formed in the body of the valve itself, the steam entering these passages at the sides of the slide-valve. The inner steam ports are at the face of the valve in the passages A A, and D D D D are the steam ports in the cylinder, two of which lead to each end; E is the exhaust port in the cylinder leading to the condenser or reservoir, and B B B the exhaust passages in the slide-valve. The exhaust steam from the outer ports reaches the exhaust cavity in the cylinder by passing through a triangular exhaust passage formed in the valve, a section through one-half of this triangular exhaust passage being marked B in Fig. 154. In the cases in which the cylinder has three steam ports at each end, the travel of the valve is still further reduced for a given area of opening.

Length of the eccentric rod.—For theoretical purposes the length of the eccentric rod is understood to mean the length measured from the centre of the eccentric strap to the centre line of the link. This must clearly be equal to the distance from the centre of the crank-shaft to the centre of the pin at the end of the slide-rod, when the valve is in the middle of its stroke.

Position of eccentric.—If the motion of the valve be considered, it will be clearly seen that its half-travel, or the throw of the eccentric, is equal to the lap of the valve, added to the maximum opening of the port to steam, which represents the extreme distance that the valve moves from its central position in either direction. Slide-valves are usually arranged so as to only partially open the cylinder port to steam, say two-thirds to three-fourths, but to open it wide to exhaust.

The position of the eccentric is best ascertained by the slide-valve diagram shown in Fig. 149, from which the most suitable

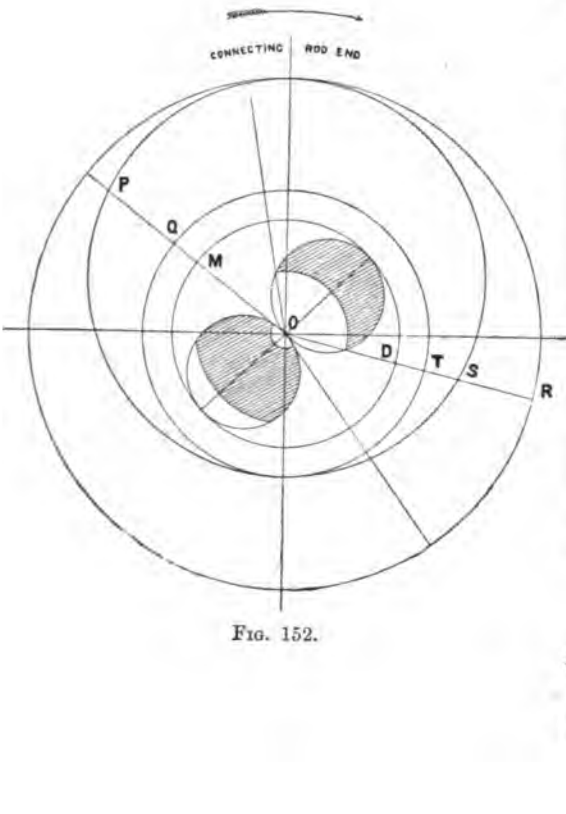


Fig. 152.

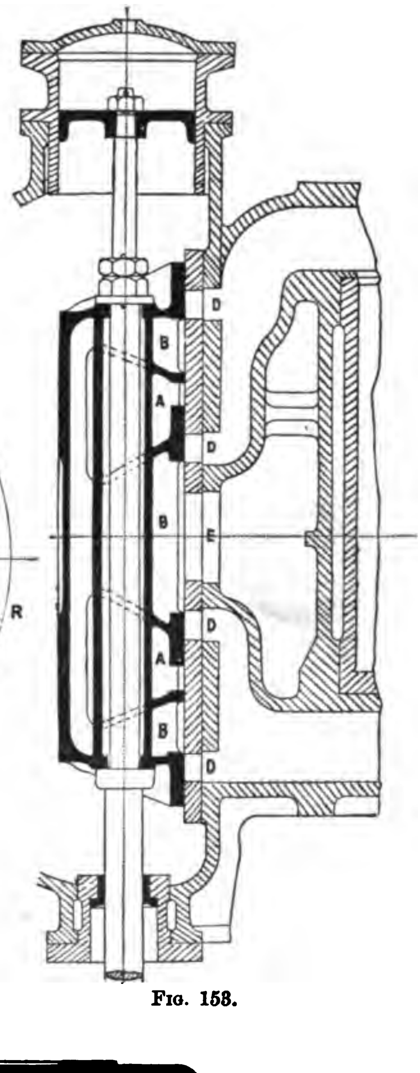


Fig. 153.

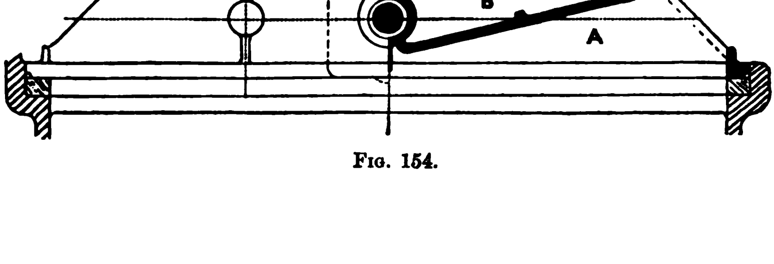


Fig. 154.

angle of advance can be determined with exactness. The approximate position is, however, frequently ascertained practically by drawing on the end of the shaft a circle with the throw of the eccentric as radius, and setting off from its centre, distances equal to the lap and required lead of the valve, as shown in Fig. 146.

Setting slide-valves.—The fixing of the slide-valve in its proper position on the rod, to insure the correct distribution of the steam, is comparatively simple, but the efficient working of the engine materially depends on it. This important operation is called *setting the slide-valve*, and for a new engine is performed thus:—

Four things have to be checked: (I.) Equality of the lengths of the eccentric rods; (II.) Equality of the radii of eccentricity; (III.) Length of the slide rod; (IV.) Amount of the angle of advance.

For I. and II. put the eccentrics on the shaft as nearly as possible in their correct positions as shown by the drawings, and secure them by the set screws. Put the link in the extreme ahead position and turn the engines ahead and note the extreme positions of the valve; do this for the astern position of the link and astern direction of engines. If the two top and also the two bottom positions of the valve agree, the rods are of the same length, and the eccentrics have the same throw, which should be correct to drawing. If they do not agree, measure the travel for both ahead and astern, if the travels are not equal the radii of eccentricity are not equal, and new eccentrics having radii as required to give the valve its correct travel, must be fitted. If the travels are equal, and correct to drawing, the lengths of eccentric rods must be adjusted till the extreme positions of the valve agree.

For III. having ascertained by actual measurement that the valve and valve face are correct to drawing, turn the engines and mark the positions of each edge of the valve at the ends of its travel, on the slide face. From these we can ascertain the lap and maximum opening at each end, if the amounts are as required the valve rod is of suitable length. If the lap at one end is too small and the maximum opening at the same end too great and the reverse at the other end, the position of the valve on the rod should be modified to correct the error.

For IV. put the engine on a dead centre, and turn the eccentric sheave on the shaft until the lead of the valve corresponds to the position of the piston at that end of the stroke; it will then be found on turning round the engines that the operations of the valve occur in their correct sequence, and that the laps and openings are correct, provided that I. II. and III. have been carried out correctly.

If it be desired to increase or decrease the lead at *both ends* of the cylinder at the same time, this must be done by the alteration of the angular advance of the eccentric on the shaft, and not by interfering with the position of the slide-valve.

In vertical engines, for reasons previously explained, the lead at the lower end of the valve is generally made greater than that at the upper side, and more exhaust lap is allowed. In this case, the valve is correctly set on the rod, when the difference between the two leads corresponds with the designed amount. Should the lead be too great or too small at both ends, the eccentric must be moved to alter this.

Relief packing-rings for flat slide-valves.—The pressure of the steam at the back of a large flat slide-valve will, unless special

provision be made to prevent it, cause great friction between the working faces of the valve and cylinder, and bring severe stresses on the working parts of the slide-gear, probably grinding the faces themselves and quickly wearing them away. To lessen the pressure between the working faces, and thus prevent these injurious results, relief or equilibrium rings are fitted for the backs of the valves, as shown in Fig. 155. These rings are sometimes fitted on the back of the valve, but are generally fitted on the slide-casing cover, and are pressed out by the action of suitable springs, so as to work steamtight on a true surface planed either on the back of the slide-valve or on the inside of the slide-jacket cover, depending on the situation of the ring, thus reducing the area on which the steam pressure can act. The space inside the packing-ring is connected to the condenser or the receiver of the succeeding engine, so that, if any leakage of steam should occur, it would not accumulate and produce pressure on the back of the valve inside the relief ring. In triple expansion engines the back of the low-pressure slide-valve is always placed in communication with the condenser, the back of the intermediate valve, if flat, being connected either with the low-pressure receiver or the condenser depending on the area of ring fitted. When flat valves are used for the high-pressure cylinder, its connection is made to one of the receivers.

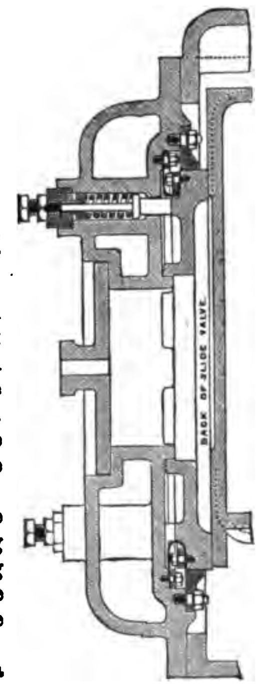


FIG. 155.

These relief rings remove a considerable pressure from the slide-valves, the area being never made large enough to prevent there being always sufficient excess of pressure to keep the slide-valve steamtight on the cylinder face under ordinary working conditions.

The connection should be made by leading a pipe from the slide-cover, in the space enclosed by the relief ring, to the receiver or condenser, as the case may be; this being preferable to drilling a hole in the back of the slide-valve to connect the hollow space inside the packing-ring with the exhaust port of the cylinder.

A cock should be provided in the pipe leading from the back of the slide-valve, and a small cock to open the space inside the ring to the atmosphere. By this means the efficiency of the relief arrangements can be tested by closing the cock in the pipe and noticing the amount of leakage as shown by the cock in connection with the atmosphere. A pressure-gauge should always be fitted to indicate the pressure inside the relief ring, so that the difference in pressure between this space and that of the slide casing can be always seen. For high steam pressures special care is necessary in this respect.

The relief rings are now generally fitted in the slide-cover, the back of the valve itself being a plane surface on which the ring slides, which arrangement possesses the advantage of enabling the packing-

rings to be adjusted at any time from the outside, whether the engines are working or not, which cannot be done if the ring is fitted on the slide-valve.

Details of various relief rings.—The details of these relief rings vary considerably, but the principle is the same in all.

Fig. 155 shows the plan adopted by Messrs. Humphrys, Tennant & Co., in which steam is prevented from passing the back of the relief ring by means of a copper spring ring, which keeps the back of the relief ring steamtight. Details of this ring are shown on an enlarged scale in Fig. 156. Stops are fitted at each end of the ring to prevent the

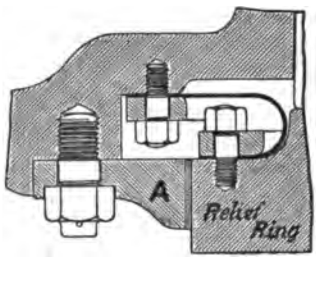


Fig. 156.

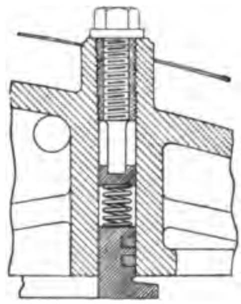


Fig. 157.

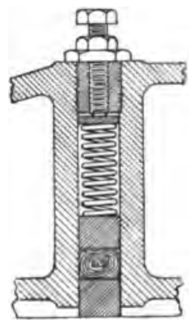


Fig. 158.

friction of the valve on the relief ring straining the copper ring. These stops are shown at A.

Other plans of relief ring are also shown, in which different methods of preventing leakage past the back of the relief ring are used. In Fig. 157 this consists of small *Ramsbottom rings.* In Fig. 158 a turn of soft packing is used as shown. Fig. 159 shows a ring working on the valve casing known as *Church's relief ring.*

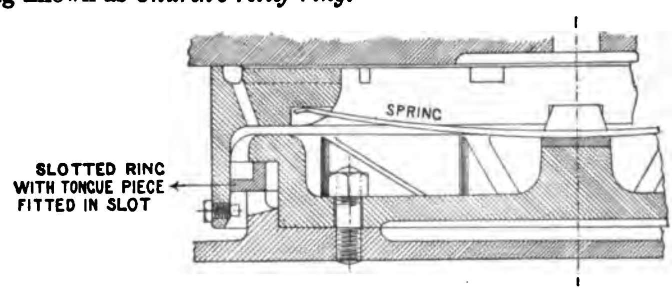


Fig. 159.

A series of spiral springs is generally fitted in recesses, pressing the relief ring against the back of the slide-valve. In two or more of these recesses the springs are omitted and stop pins fitted, a sketch of one being given in Fig. 160. These are so fixed that their points are about $\frac{1}{32}$ inch clear of the relief ring, and they prevent the slide-valve from leaving the cylinder face by more than this distance. A washer should be fitted, so that these cannot be inadvertently screwed up beyond their

correct positions. In Fig. 159, showing one form of Church's ring, a flat spring is used for pressing it against the valve-casing cover.

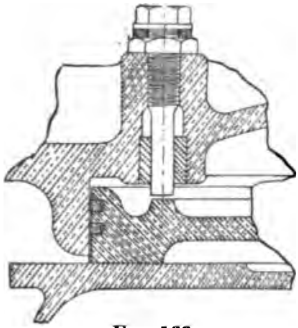


Fig. 160.

Unfortunately, however, relief rings for the back of flat slide-valves do not behave in an entirely satisfactory manner. They are troublesome to make efficient when new, and are difficult of accurate adjustment, and it is often found that they do not remain efficient for long periods, especially with the higher pressures.

Piston slide-valves. To overcome this defect, the slide-valves of the high and usually the intermediate cylinders of engines using high-pressure steam are fitted with cylindrical or piston slide-valves, instead of flat slide-valves, so that the steam does not cause any pressure between the rubbing faces, and no relief arrangements are necessary. The valve is formed by two pistons connected together, which work in cylindrical chambers that contain the steam ports, and are generally kept steam-tight by spring rings in the usual manner. The face of each of the pistons corresponds to the bars of the single ported flat slide-valve, and is of the same length, i.e. it is just long enough to cover the steam ports and allow the necessary lap.

These valves are single-ported, as double-ported piston-valves

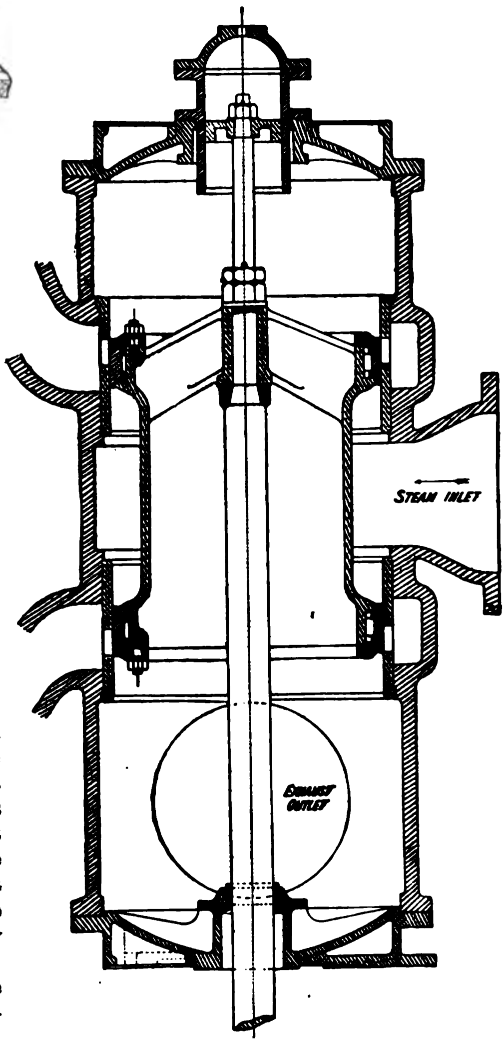


Fig. 161.

would be very complicated, and are unnecessary, owing to the considerable length of port secured by the cylindrical form.

The face of the circular slide-valve may be imagined to be formed from a flat valve face by curving the latter into the circular form. The steam is either admitted to the spaces on the outside of the two pistons with the exhaust space between them, or *vice versâ*. In the form shown in Fig. 161 the steam is admitted to the space between the two pistons around the tube that connects them together, while the exhaust takes place at the outer ends of the pistons, and the exhaust steam from the opposite ends of the valve-chest is in communication through the tube that connects the two pistons together. In Fig. 162, which shows the lower half of a piston-valve at the centre of its stroke, a steam pipe connecting the two ends is carried outside the cylinder. The steam is admitted to the cylinder from the outsides of the valves, the exhaust taking place on the inside. The steam ports are formed in separate cylindrical faces secured to the cylinder ports, as shown, and the openings are stayed with bars, that run diagonally across and serve also as guides for the piston-valves, and prevent the packing-rings springing out into the ports.

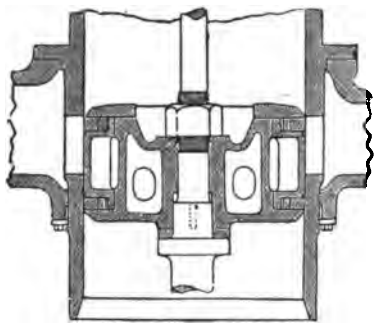


FIG. 162.

Balance pistons, &c.—In vertical engines the weight of the slide-valve, rod, and link-gear will all be taken by the eccentric and its strap, unless means are fitted to prevent this. With large slide-valves in vertical engines balance pistons are therefore fitted to take the weight of valves and gear off the links and eccentrics. Small cylinders with pistons are fitted on top of the slide casing immediately over the slide-valve, as shown in Figs. 153 and 161. The lower side of the balance piston is in connection with the slide casing, the steam pressure in which acts on the balance piston, the area of which is so arranged that the total steam pressure is sufficient to balance the weight of the gear.

The top of the balance cylinder is connected by a pipe with the exhaust steam from the cylinder.

With piston slide-valves the balancing can be effected by making the diameter of one end of the valve a little greater than that of the other, so that the steam pressure acting on the excess area balances the weight of valves and gear.

Momentum cylinder.—With very fast-running engines the momentum of the moving slide-valves brings considerable forces to bear on the eccentrics and link motion, so that it is important to reduce the weight of the valves to the lowest point consistent with strength. In fast-running engines of torpedo boats and destroyers the valves are often of gunmetal, or similar composition, to effect this object ; but even when all is done that can be in this direction, the momentum forces at speeds of, say, 400 revolutions per minute are very considerable. A simple momentum cylinder is often fitted above the slide-valve to neutralise this. This consists of a small piston and cylinder, with arrangements

for compressing a certain volume of steam or air at the end of each stroke of the valve, to absorb the forces due to the velocity of the valves and bring them gradually to rest without strain.

Joy's assistant cylinder.—Sometimes arrangements are supplied with vertical slide-valves, not only to support the weight of the valve, but to also relieve the eccentrics and link-gear of most of the work required to move the valve to and fro. One such arrangement is Joy's assistant cylinder, Fig. 163.

This consists of a small cylinder and steam-piston attached to the valve-spindle. The cylinder has a central inlet for steam, A, and two exhaust ports, B, one for each end, leading to a common exhaust pipe, and the piston is so constructed that by its motion the operations of steam admission, cut-off, release, and compression are performed on each side of the piston. The apparatus is, therefore, a small engine which exercises a force on the valve to move it up or down, and cushions steam at each end to absorb the momentum forces. These assistant cylinders give diagrams similar to that of an ordinary engine; they exert from 15 to 25 I.H.P. each for the sizes fitted in marine engines, and the amount of power developed can be adjusted by means of a valve on the steam-pipe. If the main valve be linked in, the assistant cylinder is also automatically similarly affected.

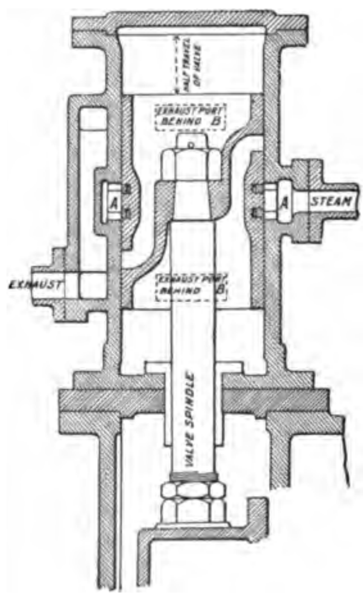


FIG. 163.

Eccentrics and rods for large marine engines.—Eccentric sheaves for large marine engines are of cast-iron, and are generally made in two parts, owing to the couplings, &c., preventing their being put on the shaft in one piece. The parts are firmly secured together by bolts. The rod is of wrought-iron or steel, and generally has a T end, by means of which it is secured to the eccentric strap by studs and nuts.

The two halves of the eccentric strap were for many years made of gunmetal. In some examples the eccentric strap, as well as the rod, is made of wrought-iron or steel, one-half of the strap being forged solid with the rod, with gunmetal liners fitted to the strap to form the rubbing surfaces.

In the most recent practice the strap is of forged or cast-steel, lined with white metal, which forms the rubbing surface working on the cast-iron sheave. This combination of metals has been found to give excellent results, and it is now specified by the Admiralty.

Fig. 164 shows the details of a modern eccentric sheave, strap, and

rod, the strap being lined with white metal. In some examples, in order to reduce the diameter of the eccentric sheave, the smaller of the two parts is made of wrought-steel, which enables the least thickness

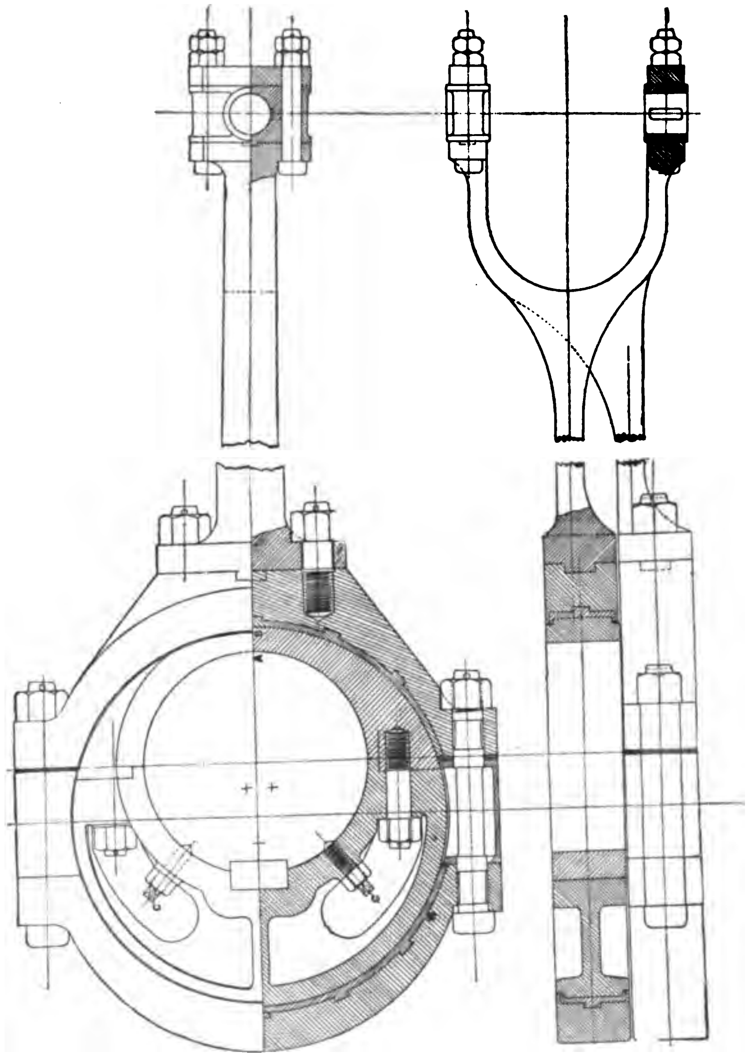


FIG. 164.

of the sheave around the shaft at A B to be diminished. Set screws, C, are fitted for purposes of adjustment before the key is fitted.

CHAPTER XVII.

STARTING AND REVERSING ARRANGEMENTS.

All marine engines must be arranged so as to be capable of being worked in opposite directions, in order that the ship may be driven either ahead or astern. It is necessary, therefore, that suitable reversing gear should be fitted to enable the slide-valves to be placed in the proper positions to produce revolution of the crank-shaft in either direction.

Loose eccentric.—In the earlier paddle-wheel steamers this was accomplished by means of a single eccentric, fitted *loosely* on the shaft,

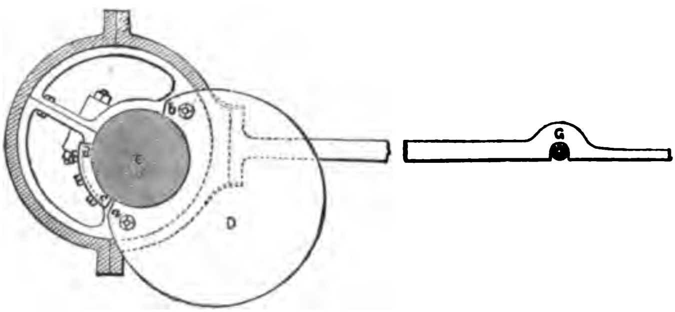


Fig. 165.

and driven by stops fixed in suitable positions on it, to give either ahead or astern motion as required. A sketch of this arrangement is given in Fig. 165.

This loose eccentric is balanced by means of the disc D, to prevent it falling away from its position when the slide-valve is moved by hand. The eccentric rod is attached to the slide-valve rod by means of the gab G at its end, which fits over a corresponding pin on the end of the slide-rod. In starting or reversing these engines the gab G is disconnected from the slide-rod, and the slide-valves worked by hand to start the crank-shaft revolving in the proper direction, or reverse its motion as the case may be, and cause the stop for the proper motion to come in contact with the eccentric, and drive it in the required direction. The gab end then drops over the pin and continues the motion.

In Fig. 166, let O R represent the position of the crank on the dead point. Then, from what has been previously explained about the motion of the slide-valve, if steam be taken at the outside edges of the valve, and the eccentric radius be in the position O P, the crank

will revolve in the direction A B C, whilst if the eccentric arm be in the position shown by the dotted line O Q, the motion of the crank will be in the opposite direction A Q C. The stops on the crank-shaft must, therefore, be so arranged as to bring the eccentric pulley to the position O P, or to the position O Q, according as the engine is to be driven ahead or astern.

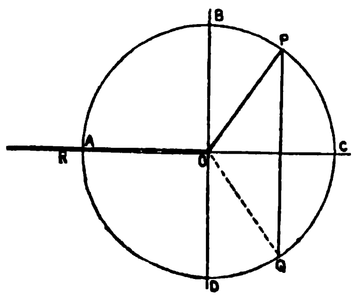


FIG. 166.

The manner in which this is accomplished is shown in Fig. 165. On the crank-shaft, C, is fixed a stop *c d*, extending a sufficient distance round its circumference, and the ends of D, the balancing disc on the eccentric, are arranged to form corresponding stops, *a* and *b*. When, therefore, the top of the shaft revolves from right to left, the edge *c* comes in contact with *a*, and the eccentric and shaft revolve together so long as the motion continues in this direction ; but as soon as the engine is reversed, the stops become detached, and after about a quarter revolution the edge *d* comes in contact with *b*, and the engine works in the reverse direction.

Link motion.—The reversing of modern marine engines is usually effected by means of the 'link motion,' which was invented by Stephenson. It is simple in construction, and not only can the engines be reversed by it, but it provides for a considerable range of expansive working of the steam, without the interposition of any other gear.

The general construction and arrangement of this gear are shown in Fig. 167. On the crank-shaft C there are keyed two eccentrics, one

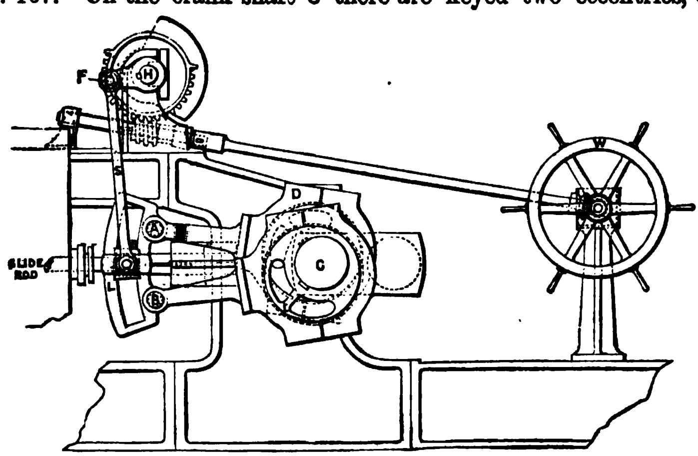


FIG. 167.

in the position to give ahead motion and the other in the position for astern motion. The eccentric rods are of equal length, and their ends are attached by working joints to the opposite ends of a curved link L.

with a slot in the centre, which slides over a brass block B, to which the end of the slide-valve rod is attached. This gear forms a ready means of throwing either of the eccentrics into gear as desired, and at the same time throwing the other out of gear.

It will be seen that when the link is moved from its middle position, so as to bring the pin A in a line with the slide-valve rod, the motion of the valve will be governed by the eccentric D, while the other simply swings the link without affecting the motion of the slide-rod. If, on the other hand, the link be moved to the other side, so as to bring the end of the eccentric rod B in a line with the slide-rod, the eccentric E will govern the motion of the slide-valve, the other eccentric having no effect, and the engine will work in the reverse direction. When the link is in the central position the motion of the valve is considerably reduced, and the distribution of the steam is such that no revolution of the engine could ensue.

The centre of the link is commonly called its 'dead point.' If the link be placed in such a position that the sliding block B is between the centre and the ahead end of the link, the ahead eccentric exercises the greatest influence over the motion of the valve, so that the engine continues to work in the ahead direction; but the astern eccentric has now some effect in modifying the motion of the valve, the result being that the travel of the slide-valve will be less than when the link is at its extreme position, and all the operations of the valve will be earlier than when in full gear, as if the valve were now being worked by an eccentric with greater angle of advance and smaller throw. The steam is therefore cut off earlier and worked expansively.

The operation of working expansively by means of the link motion is technically called 'linking up' in horizontal engines, or 'linking in' for vertical engines, or, generally, 'shortening the link.'

Varieties of links. — There are three varieties of links used: (*a*) the slotted link, (*b*) the solid-bar link, and (*c*) the double-bar link.

The *slotted link* consists of a curved bar with a slot cut in it (Fig. 168), in which slot the link block is fitted. This link block is attached to the slide-rod by a pin about which an oscillating motion of the block occurs. Two projections are formed on the link on one side, with eyes to which the ends of the two eccentric rods are attached. This is the original form of Stephenson's link, and is still commonly used in small engines. The centre of the eccentric-rod end in this form does not usually coincide with the centre line of the link block, so that the motion is not so regular or the means of adjustment so good as in plan (*c*), which is now the general plan used for large vertical engines.

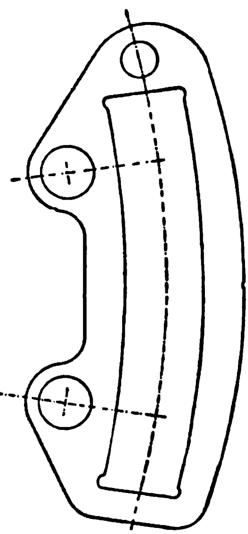


Fig. 168.

The *solid-bar link* (Figs. 169 and 170) is always fitted by Messrs. Humphrys, Tennant & Co., and consists of a simple curved rectangular

bar, with eyes formed at each end for the attachment of the eccentric rods. The solid bar passes through the block, which consists of two segmental pieces of gunmetal, cylindrical on the outside, and which

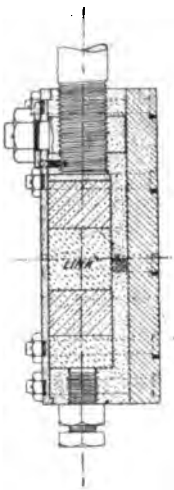


FIG. 169.

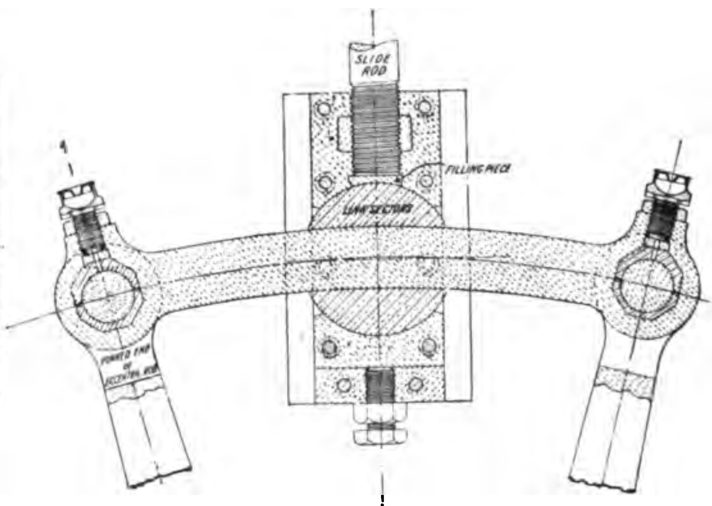


FIG. 170.

have an oscillating motion in its bearing when at work. This variety also has the advantage of being easily adjusted in all parts. The end of the eccentric rod cannot be brought in line with the slide-rod, and even in full gear the motion is one with a 'shortened link.' The sketches show in detail the construction and means of adjustment of all parts.

The most general plan is the *double-bar link* (Figs. 171 and 172) consisting of a pair of curved steel bars joined together at the ends,

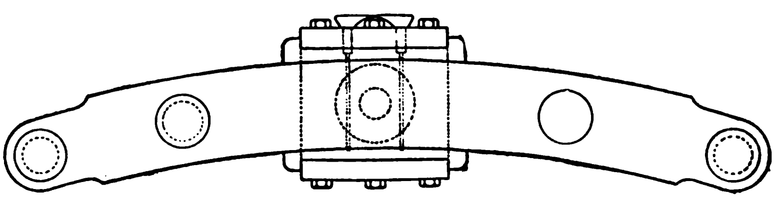


FIG. 171.

and kept a certain distance apart by distance pieces. Projecting pins are formed on the link bars, two on each side, for the attachment of the eccentric rods. The ends of the eccentric rods are forked, and contain each two adjustable bearings which embrace the pins on each side of the link. The link block consists of a steel or iron pin sliding between the bars, with top and bottom projections each side, which embrace the link bars. The link bars slide through these projections, and adjustable gunmetal liners are fitted as working surfaces between

the link block projections and the link bars. All parts are capable of ready adjustment, and when the link is in full gear the centre of the

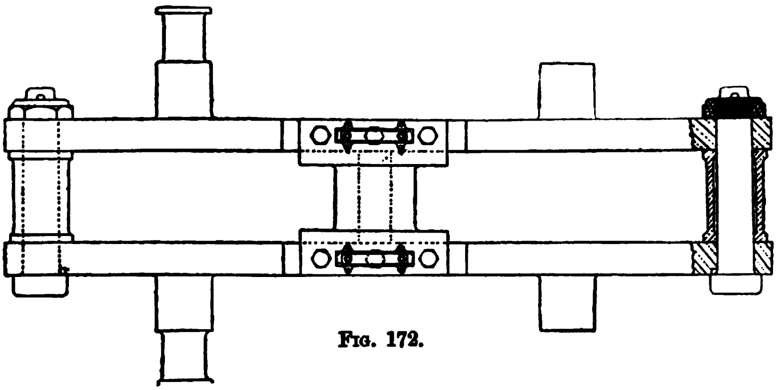


FIG. 172.

eccentric rod end coincides with that of the link block. Detailed sketches of the link block and its lubrication arrangements are given in Figs. 173 and 174.

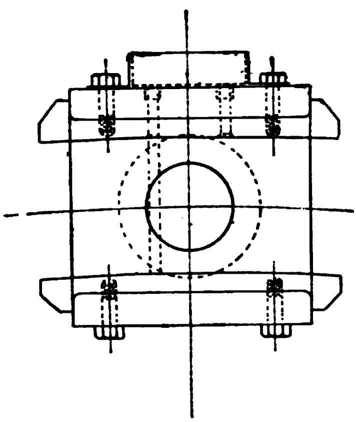


FIG. 173.

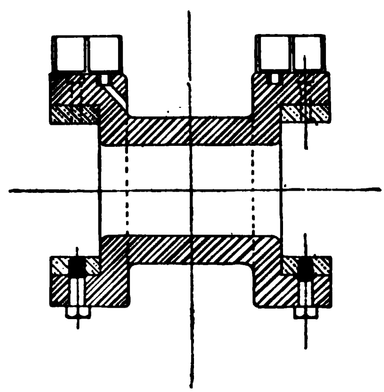


FIG. 174.

These different arrangements of link motion, however, only differ in the details of construction, the principles involved being the same in all.

For detailed construction of the eccentric sheave, strap, and rod see end of preceding chapter.

Curvature of link.—If the link were used simply for reversing the engines, its amount of curvature would not be of importance, or it may even be straight; but as it is required to be used for working expansively, its shape must be such that when the block is in any intermediate position the centre of the travel of the valve will be always constant, otherwise the distribution of steam between the two ends of the cylinder would be interfered with. To effect this quite accurately the

link should be a parabola of large focal distance, but an approximation sufficiently close for practical purposes is obtained by making it a circular arc of radius equal to the length of eccentric rod—i.e. the length between centre of eccentric sheave and centre of pin at end of rod.

Open rods and crossed rods.—If the eccentric rods are so placed that when the crank is pointing *towards* the link in cases where steam is taken on the *inside* edges of the valve, and pointing *away* from the link in cases where steam is taken on the *outside* edges of the valve, the rods are not crossed, the gear is said to have '*open rods.*' If, when the crank is so placed, the rods cross one another, the gear is said to have '*crossed rods.*' Figs. 175 and 176 show open and crossed rods respectively for the case in which steam is taken on the outside edges of the valve.

The motion produced on the slide-valve when the link block E is at any intermediate position, F, of the link may be found geometrically as follows. Connect the points A and B by an arc of a circle of radius $= \frac{1}{2}\frac{AB}{CD}\cdot AC$. For open rods this arc should be concave, and with crossed rods convex, to the centre O. If this arc be divided at the point G, in the same ratio that the point F divides the link, the motion of the slide-valve will be very nearly the same as if it were worked, direct, by an eccentric arm O G, having an angle of advance equal to the

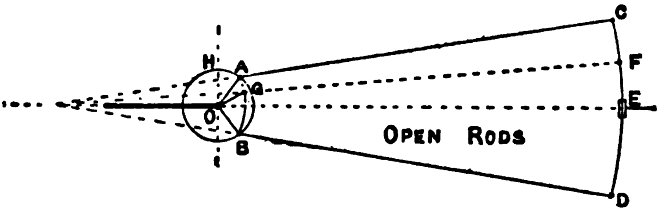


FIG. 175.

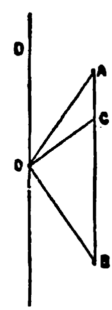


FIG. 177.

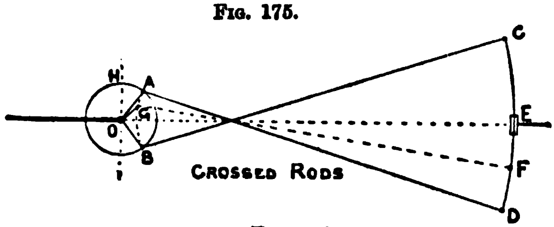


FIG. 176.

angle H O G. O G is called the '*virtual eccentric radius,*' and H O G the '*virtual angle of advance.*'

A simple approximation to the motion of the valve for any position of the block in the link, when the rods are long, may be made as follows. Let O, Fig. 177, represent the centre of the shaft, O A the ahead, and O B the astern eccentric radius. In full gear, O A is the throw of the eccentric, and the angle D O A is equal to the angular advance. For any intermediate position, divide the line A B at C in the same ratio that the block divides the link; then the motion of the valve will be due, approximately, to an eccentric arm O C, set at an angle of advance equal to the angle D O C.

By drawing Zeuner's valve circle for the linked-in positions we see that all the operations are earlier, also that with open rods the lead greatly increases as we link in, while with crossed rods it slightly diminishes and may become negative. The objection to crossed rods is that the travel diminishes so considerably when linking up, causing wire-drawing, and for this reason they are not so common as open rods.

Starting gear.—The link is suspended generally at its centre or ahead end, as shown in Figs. 178 and 167, by the suspending rod S; and during the working of the engines it oscillates about the pin F at the end of the suspending rod.

The object of the starting gear is to move the link into the proper position, to bring the correct eccentric into action for the required

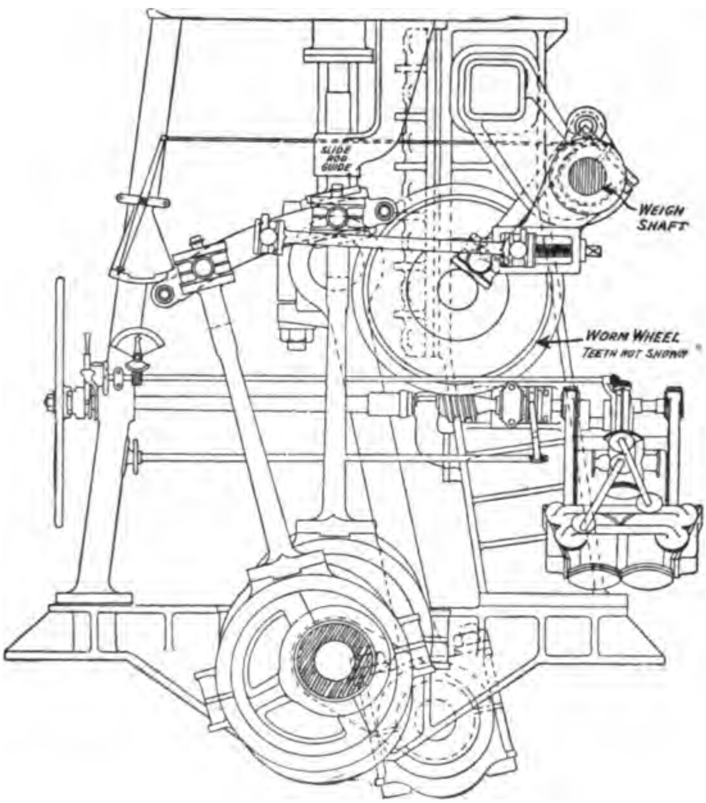


FIG. 178.

motion, and this is done in small engines, in which little force is required to move the slide-valves, by means of a simple lever.

In engines in which the slide-valves are large and require considerable force to move them, the starting gear is worked by steam as well as by manual power. The use of steam saves labour and considerably facilitates the handling of the engines. Steam-starting

gear is now generally fitted to all marine engines, except very small ones. This also prevents the necessity of crowding the starting platform with men to stand by the starting gear, and renders it possible for the engines to be reversed quickly in case of emergency, with only the engineer of the watch in the engine room, without waiting for the assistance necessary with hand-starting gear.

Steam-starting gear fitted in H.M. Navy.—Fig. 178 shows a general arrangement of starting gear of the type used in the Royal Navy. The reversing and starting arrangement consists of a weigh-shaft running along the upper part of the engine columns on which are keyed a series of reversing levers, one to each link, which are attached to the latter by suspension rods, so that the operation of reversing or starting consists in moving this weigh-shaft through a certain angle. An additional lever is fixed on the weigh-shaft near the starting engine, and by means of this lever the starting engine actuates the weigh-shaft and moves the links to and fro. The engine is generally an ordinary double-cylinder reversible steam-engine which works a worm geared to a worm-wheel. To a point in this worm-wheel, the lever above referred to is attached by means of a rod. The angle through which the reversing lever and weigh-shaft turns between the extreme positions of ahead and astern is clearly governed by the diameter of the circle traversed by the pin on the worm-wheel, and the proportions are so arranged that the extreme travel is only just sufficient to move the links to the required positions. The worm-wheel is capable of continuous circular motion, and if the reversing engine is allowed to travel beyond the proper position no harm is done, and the links are only brought back again a small distance.

This is often spoken of as an 'all round' reversing gear, and with it the starting engines were often made non-reversible, and the worm-wheel allowed to travel round until the required position is reached. A longer time is thus taken to manipulate the engines. In the Royal Navy, however, where quick manœuvring is essential, they should always be reversible. This continuous motion reversing gear is also very useful for safely and quickly warming up the engines, for as soon as steam is available, if the reversing engine be kept slowly revolving in the same direction, the links are kept moving up and down and a small amount of steam passes safely through the engines.

To reduce the rapidity of the motion a double worm arrangement is often fitted, in which the engine is not connected directly to the main worm-shaft, but to a smaller worm, working a worm-wheel fixed on the main worm-shaft.

A hand wheel is fitted for use when steam is not available, or should any accident derange the starting engine, two clutches being fitted as shown, so that either the hand wheel or the starting engine can be used independently.

In cases in which a rotary engine is employed for starting purposes and the 'all round' arrangement is not fitted, there is a possibility of the engine running beyond the extreme working position, and either jamming the screw or other gear used, or straining the link motion. In such cases automatic stopping gear is necessary for the starting engine.

The steam for working the starting engine should be taken from a branch on the main steam pipe and not from the auxiliary steam

service, the branch being on the boiler side of the regulating valve of the main engines.

Mercantile marine starting engine.—The general type of starting

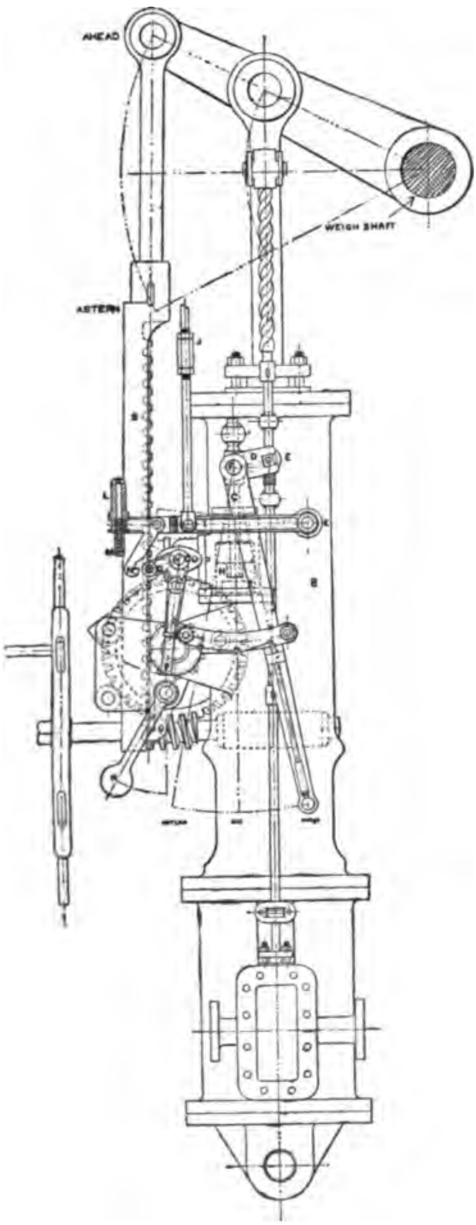


FIG. 179.

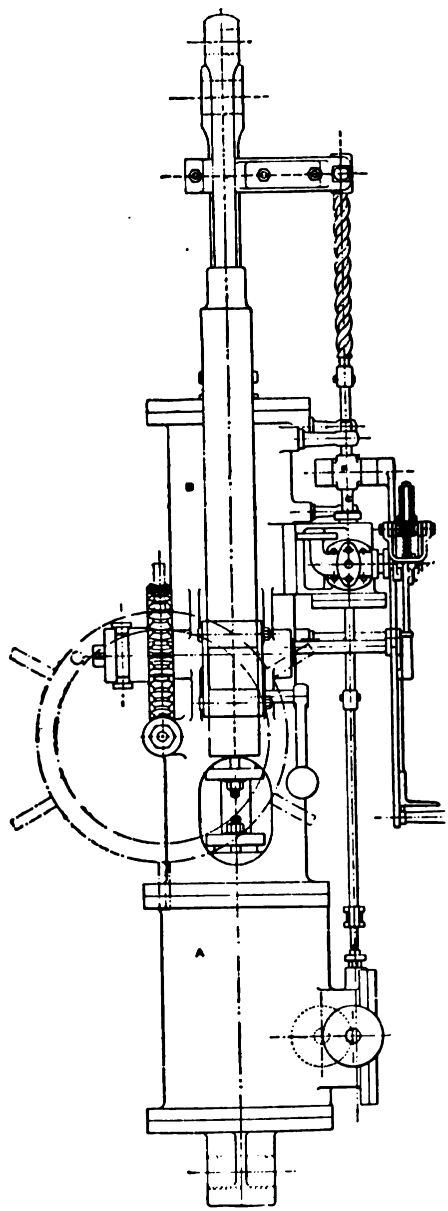


FIG. 180.

engine fitted in the mercantile marine consists of a direct-acting steam-engine with hydraulic cylinder brake, the piston-rod of which acts on the end of the weigh-shaft lever, pulling it to and fro, and thus moving the links and stopping or reversing the engine.

One of the most extensively used of such gears is that designed by Messrs. Brown Bros. & Co. Sketches of this reversing gear, similar in principle to that supplied by this firm to many vessels, are given in Figs. 179 and 180.

The reversing engine is attached to the bed-plate or column of the marine engine by the oscillating joint formed on the lower end of the steam cylinder A. In this cylinder a piston and rod are fitted, the latter being continued through the top and attached to a metallic-packed piston, working in the hydraulic cataract cylinder B, which is filled with water to steady the motion of the engine. The rod passes through a metallic-packed stuffing-box on the steam cylinder, and through stuffing-boxes at each end of the hydraulic cylinder, and its upper extremity is attached to the weigh-shaft lever which actuates the links of the main engine.

The starting engine is handled by the long lever indicated, working in a quadrant, which is notched for the positions ahead or astern, or any intermediate expansion necessary. On moving this lever in either direction it moves the valve-rod of the steam cylinder A, and admits steam to one side of it, and thus actuates the weigh-shaft and the link motion. A horizontal arm with coarse spiral nut is attached to the piston-rod, working on a coarse spiral thread cut on a prolongation of the valve spindle of the cylinder A, the motion of the reversing engine, with the arm and spiral-nut, causes the valve spindle to revolve, screwing it back to the shut position through the nut E (Fig. 179), which is operated upon by the reversing lever.

The oscillation of the engine is so small that any practical length of copper pipe for steam or exhaust usually met with in marine engines is sufficient to give the requisite amount of elasticity without stuffing-boxes.

The independent hand gear with locking arrangement is also shown in the figures. The locking device Q is fitted with a balance weight R, and engages the teeth of a rack. When working with shortened links, the reversing valve is left slightly open to keep a strain upon the pawl, so that the links are held firmly. At the same time the engine is ready for immediate reversal astern. Any motion in that direction causes the pawl to fall out. The rack S is actuated in the usual way by a pinion attached to a worm-wheel shaft, which in its turn is operated upon by the worm and wheel driven by the hand wheel as shown.

In many recent engines Messrs. Brown's automatic emergency governor has been fitted, in view of some serious casualties which have occurred through the racing of marine engines to a dangerous extent, caused by broken shafts or propellers.

Brown's governor gear.—The governor arrangement is shown in the drawings, and acts as follows :—

The reversing lever is in this case attached to a *movable* fulcrum F attached to a piston working in the small additional steam cylinder H. This fulcrum is *fixed* in cases where no governor gear is fitted. The

cylinder is connected to the cock O, which admits steam to the bottom of the piston when the engines are working at safe speed. The end of the reversing lever D works the starting engine valve gear at E. Another lever I is reciprocated about three inches by the rod J, attached to the indicator gear, air-pump levers, or other parts of the engines. This lever works on a fixed fulcrum K, and carries a small weight L supported on a spiral spring in the box M, adjusted to act at an unsafe speed. This weight has a groove into which the upper end of the bell-crank lever N gears, while the lower end is ready on emergency to engage and turn the lever of the cock O.

When the engines are working at their normal speed the spring is so set that the weight L (which compresses the spring at each stroke of the engine in virtue of its momentum) shall not cause the lower end of the lever N to approach too near the lever O. Should the engines exceed a safe speed the momentum of the weight will compress the spring, so as to cause the hook N to engage the lever on the cock, turning steam above the piston of cylinder H, and exhausting it from the bottom, pulling the fulcrum F down, carrying with it the starting-engine valve-rod E, which turns steam on the top of the piston of starting engine A, and so moves the links into or near mid-gear.

The motion of the main engines is thus arrested simultaneously in all the cylinders, more rapidly than by closing the stop-valve, however quickly the latter may be effected. When this action has taken place the piston of the governor cylinder can be returned to its normal position by the handle P, when necessary to restart the engines. As the links are only moved to mid-gear, the apparatus forms an efficient governor without completely stopping the engines, as the momentum of the ship causes them to revolve slowly, when the attention of the engineer on watch is at once attracted. Being always in motion and in sight, it is not open to the objection frequently made of governors, that they are liable to be found out of order when wanted.

Differential reversing valve.—Steam-starting engines are made reversible, sometimes by fitting them with double eccentrics and link motion, but more generally by fitting them with a special reversing valve, called a '*differential valve*,' shown in Fig. 181, which illustrates one form of starting engine fitted with automatic stopping-gear.

The engine in this case is fitted with a single fixed eccentric; and as this must be capable of working the starting engine in either direction, it must be keyed on the shaft at right angles to the crank (see page 178)—that is, the eccentric has no angular advance, and consequently the slide-valve must be without either lap or lead.

The reversing valve, or differential valve, is shown at K, and is of similar construction to an ordinary slide-valve, but it has suitably arranged ports, and is worked by hand by means of a lever. It may be either an ordinary slide-valve, as there shown, or, more generally, a cylindrical valve, the action of which, however, is precisely similar to that of a flat valve. The space A outside the reversing valve K is kept supplied with steam, while the hollow of this valve is in connection with the exhaust. The valve K, as drawn, is in its middle position, and the reversing engine is stopped. If, now, the reversing valve K be lowered by moving the handle to the right, steam flows from A through the upper port to the outer edges of the cylindrical

slide-valve, while the inner edges of this slide-valve have free communication with the exhaust through the lower port of the reversing valve, and the engine then rotates.

If the reversing valve K be now raised, the upper port of the reversing valve is cut off from the steam space A, and placed instead in connection with the exhaust through the hollow of the reversing valve, while the lower port of the reversing valve is opened to the steam supply space A, instead of being, as before, in connection with the exhaust. The inner edges of the cylindrical slide-valve are now

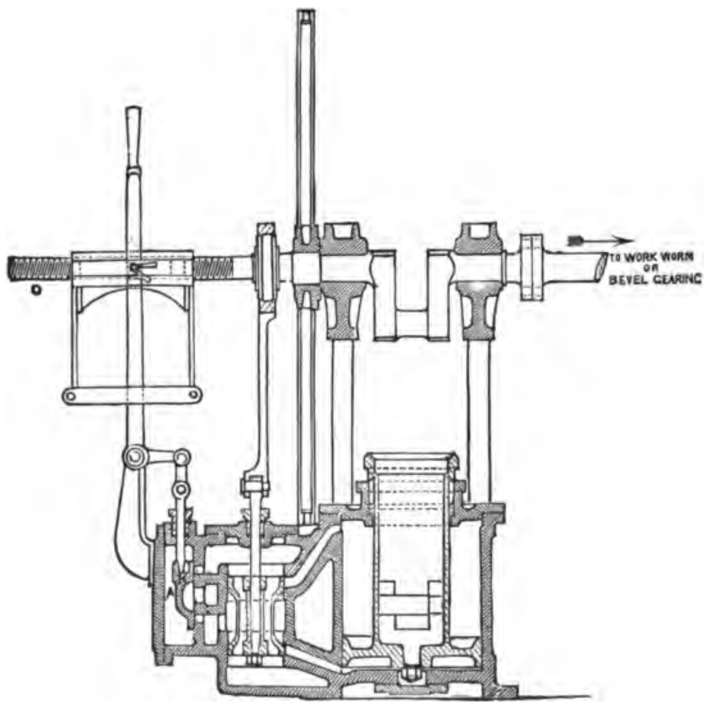


FIG. 181.

supplied with steam, while the outer edges are in connection with the exhaust, so that the steam and exhaust spaces of the engine slide-valve are interchanged, and the steam pressure is transferred from one side of the piston to the other, consequently the engine now moves in the reverse direction.

It should be observed that steam and exhaust are always on the same side of the reversing valve K, but changes at the engine slide-valve. The steam and exhaust supply to the reversing valve K are not shown on the drawing. In the particular example illustrated the shaft of the engine is screwed at D, and works a nut which actuates a frame at the reversing handle. When the latter is moved the revolution of the starting engine works the frame along, and brings the

reversing handle back again to the vertical position, thus stopping the engine.

If flat slide-valves are fitted to the engine provision must be made to prevent them from being forced off the cylinder face, when the steam pressure is acting inside the valve and the outside is connected with the exhaust. Owing to the absence of lap on the slide-valve this arrangement is not economical in the distribution of steam, but it is very convenient, and is largely adopted in small engines, such as starting, steering, turning, turret, capstan, boat-hoisting engines, &c., in which economy is not of the first importance.

Radial valve gears.—Many different arrangements of gear for working the slide-valves have been designed to supersede the link-motion. Several of them produce a good distribution of steam at all points of cut-off, and have been often used in recent engines. In most of these slide-valve gears the motion of the valve is obtained by compounding two motions, one in the direction of motion of the piston and the other at right angles to it. By suitably proportioning the various parts and adjusting their positions relative to each other, any degree of expansion may be obtained with uniform lead at all points of cut-off, or the direction of motion of the engines reversed as may be desired.

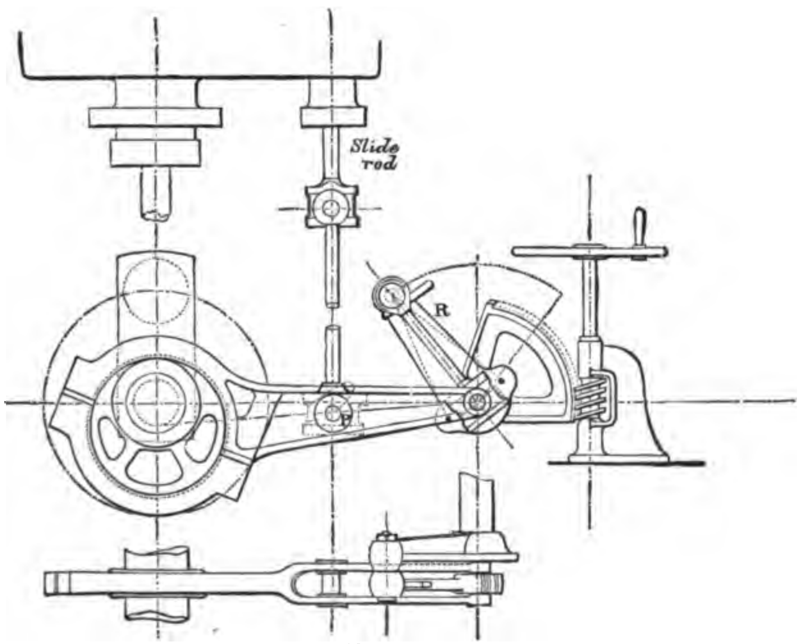


FIG. 182.

Marshall's valve gear.—This arrangement of slide-valve gear, which has been fitted by Messrs. Hawthorn, Leslie & Co. to a large number of marine engines, is illustrated in Fig. 182. In this system only one eccentric is used, the end of the eccentric rod being attached

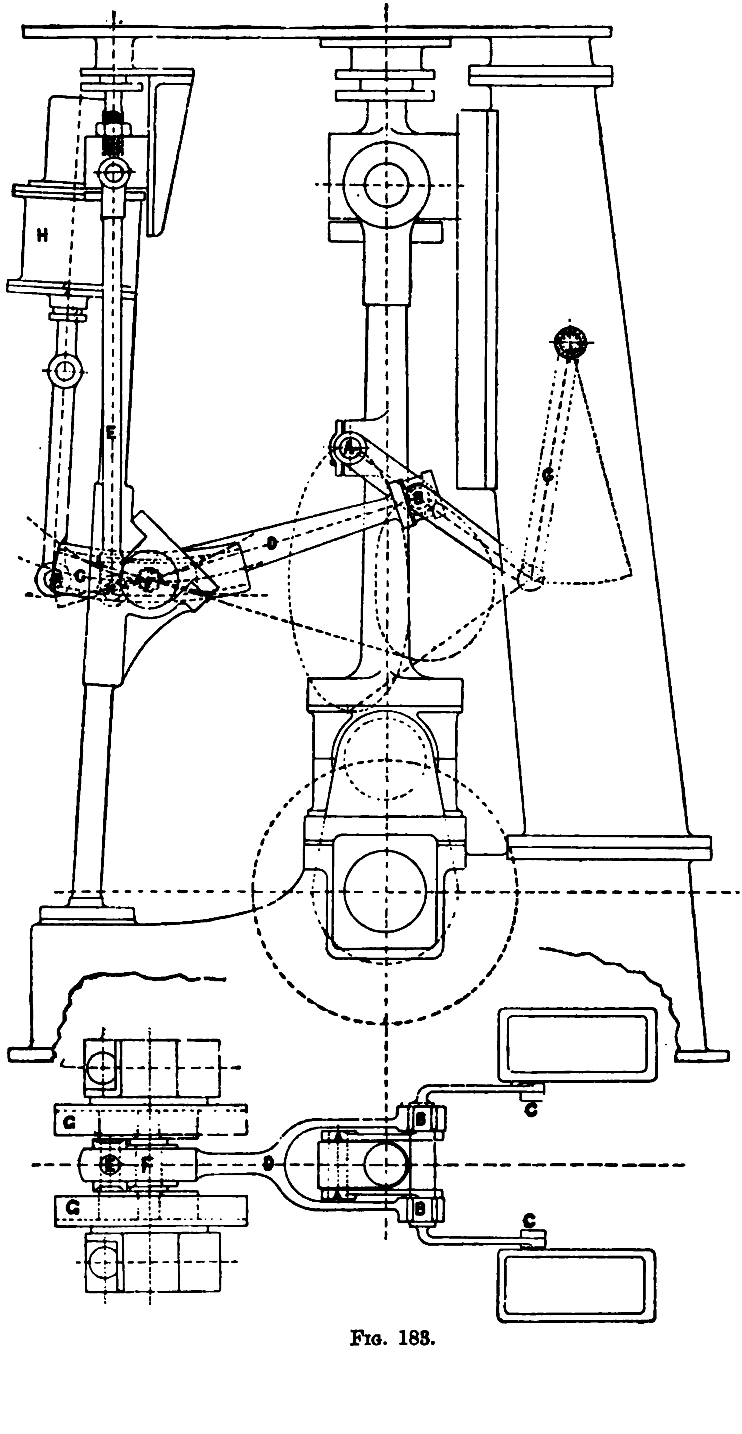


Fig. 183.

to a rod hung from a pin on the reversing shaft lever, R, by which it is constrained to move in an arc of a circle inclined to the centre line. To an intermediate point, P, in the eccentric rod a connecting link is attached which communicates the necessary motion to the slide-valve rod. By adjusting the position of the reversing lever, R, any desired degree of expansion can be obtained, or the engines reversed as required.

Joy's valve gear.—Fig. 183 shows an elevation and plan of the latest arrangement of Joy's valve gear applied to a vertical engine. In this gear eccentrics are dispensed with, and the movements of the slide-valve obtained from the connecting rod. The vibrating link B, jointed to the connecting rod at A, has one end constrained to move horizontally by the action of the radius rod C. One end of another rod, D, works on a pin in the vibrating link B, near the other end is a fulcrum carried by a pin F attached to sliding blocks on each side working in sectors G, which are carried by the reversing shaft, the centre line of the sector passing through the centre of the reversing shaft. From D the motion is communicated to the slide-valve rod by means of the link E, attached to a point K in the rod D beyond the fulcrum F.

The forward or backward movement of the engine is governed by inclining the sector on one or the other side of the horizontal centre line, and the amount of expansion depends on the amount of the inclination, the exactly central or horizontal position being 'mid-gear.' The reversing arm F R moves these sectors to the required position, and its extremity R is connected to the starting engine H. The paths of the point A in the connecting rod, and also of the point B in the vibrating link, as the engine revolves, are indicated by dotted lines, as are also the extreme positions of the sector centre lines for ahead and astern working respectively. The gear as drawn is in the stop position. By this gear a constant lead is secured for all linked-up positions, since when the piston is at the top or bottom of the stroke the pin F coincides with the centre of the reversing shaft, so that in this position any movement of the sectors does not affect the position of the slide-valve. The up and down motion of the point B therefore gives a constant movement of the valve equal to the *lap plus the lead*, while the horizontal motion sliding the block to and fro in the sectors adds the amount required for steam opening, this amount increasing with the angle of the sector to the horizontal.

This gear has been fitted to a large number of locomotive and marine engines. Where applied it will be seen that the slide-valves are in front of the engines, which shortens the length of the engines considerably. With the ordinary link motion the intervention of an intermediate shaft is necessary to work the valves in this position.

CHAPTER XVIII.

ARRANGEMENT OF THE CYLINDERS OF COMPOUND, TRIPLE, AND QUADRUPLE EXPANSION ENGINES.

THE type used for modern engines in the mercantile and Royal navies is either the triple or quadruple expansion engine, by which with high-pressure steam considerable advantage is gained over the earlier types of engines as regards economy, in reduction of heavy stresses on the machinery, and in other respects.

We will first describe the reasons for the superiority of the old compound over the simple engine, and the arrangement of cylinders in the former, as this will suggest the similar reasons which, with increased steam pressure, subsequently led to the abandonment of the compound type and the introduction of triple, and also large numbers of quadruple expansion engines. Only one quadruple expansion engine above the steam pinnace size has so far been fitted in the Navy, viz. that of No. 90 first class torpedo boat, but their use in the mercantile marine is gradually extending.

The principal difference between the mechanism of the old simple expansion engines and that of triple expansion and other stage expansion engines is in the arrangement of the cylinders, the other parts being generally the same. In the simple expansion engine the steam enters each cylinder direct from the boilers, and at the end of each stroke is exhausted direct into the condenser. In the compound engine the steam from the boilers is only admitted direct to the smaller or high-pressure cylinder, and at the end of the stroke in that cylinder, instead of passing direct to the condenser, the steam enters a larger cylinder, called the 'low-pressure cylinder,' in which the expansion is completed, after which the steam passes as before to the condenser.

As will have been gathered from Chapter XIV., the compound engine has now been generally superseded by the triple expansion engine, in which the steam from the boilers is admitted to the high-pressure cylinder, from whence it is led to a larger cylinder called the 'intermediate-pressure cylinder,' in which it expands further and performs more work. On being exhausted from the intermediate cylinder, it is conducted, as in the previous case, to a still larger cylinder, called the 'low-pressure cylinder,' in which its expansion is completed, and on being discharged from this cylinder it proceeds to the condenser.

Two-cylinder compound engines.—Two principal types of two-cylinder compound engines used to be fitted. The type shown in Fig. 184, generally known as the 'tandem' type, has certain advantages

and was largely adopted, especially for engines of large power. Two pairs of cylinders were fitted, so that large powers were obtained without introducing castings of extraordinary complexity. It was also the readiest form to which a simple engine could be converted.

This sketch shows the cylinders of a horizontal tandem compound engine with return connecting rods, a few examples of which are still at work in the Navy. In this type of engine a defect was that the clearance spaces in the high-pressure cylinders were very great, which decreased the expansive efficiency and caused considerable waste of steam.

The simpler and more usual arrangement was that with the high-

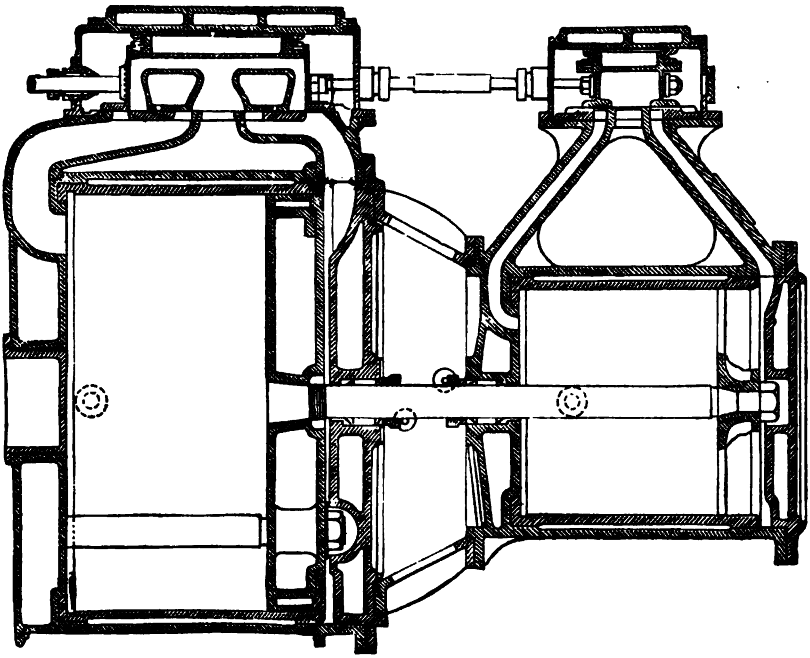


FIG. 184.

and low-pressure cylinders placed side by side, the pistons acting on cranks at right angles to each other.

Three-cylinder compound engines.—The ordinary three-cylinder compound engine is simply a modification of the type just described, but instead of a single low-pressure cylinder, two cylinders are used, the steam on exhausting from the high-pressure cylinder to the receiver being conducted to two low-pressure cylinders.

The three-cylinder type of compound engine was used when the power was so great that the employment of a single low-pressure cylinder would be inexpedient on account of its unwieldy dimensions, so that the division of the work between two low-pressure cylinders is

preferable. The angles at which the cranks of three-cylinder compound engines were placed with respect to each other were very varied. In the majority of cases they were set at equal angles of 120°, but various other arrangements of cranks were common, depending on the opinion of the makers as regards regularity of twisting moments, and distribution of steam. It is, however, doubtful if any of these variations possessed any practical advantage over that of placing the cranks at equal angles with each other.

Definition of the term 'receiver.'—By the term 'receiver' is to be understood in the case of a compound engine the whole of the space between the high-pressure piston when at the end of its stroke, and the back of the low-pressure slide-valve or valves, comprising the volumes of the steam and exhaust passages of the high-pressure cylinder, the exhaust pipes from the high-pressure cylinder to the low-pressure valve casings, and the low-pressure valve casings themselves.

In the case of a triple expansion engine, the space between the high-pressure piston at the end of its stroke and the intermediate slide-valve is called the 'intermediate receiver,' and that between the intermediate piston at the end of its stroke and the low-pressure slide-valve the 'low-pressure receiver.'

Capacity of receivers.—Large reservoirs or receivers for the steam between the cylinders were usually fitted to the first compound engines, but experience proved that they were not required, all that was found necessary being a comparatively large exhaust pipe from the eduction orifice of the high-pressure cylinder to the steam inlet of the low-pressure cylinder, the volume of the exhaust passage and pipe from the high-pressure cylinder and the low-pressure valve casing being sufficient to allow for the compression that takes place between the release from the high-pressure cylinder and admission to the low-pressure cylinder. Similar remarks apply to the receivers of triple and quadruple expansion engines, and the volumes of these spaces which are necessary for other reasons are found to be sufficient for receivers. Most modern engines are made in this way.

The capacity of the receivers is immaterial so far as the total power of the engines is concerned, its effect being shown on the back pressure line of the diagram from the preceding engine, which becomes more nearly straight, and on the admission line of the diagram from the succeeding engine, which becomes more nearly parallel, to the atmospheric line as the volume of the receiver is increased.

Influence of size of cylinder on the power of stage expansion engines.—The power of any stage expansion engine, *working at any given rate of expansion*, depends entirely on the dimensions of its low-pressure cylinders, and is not affected by the size of the high-pressure cylinder, which must only be regarded as carrying out one stage in the expansion. The capacity of the low-pressure cylinder or cylinders of such an engine requires to be the same as that of the whole of the cylinders of a simple expansion engine of the same power working at the same initial pressure of steam and total ratio of expansion. Neglecting for the moment the complicating effects of clearance and compression, this will be easily seen from the consideration that since the initial pressures and the ratios of expansion are the same, the final pressures

and volumes must be identical in the two cases. In the simple engine the whole of the steam at the end of the expansion fills all the cylinders, whilst in the compound engine it is contained by the low-pressure cylinders only. Consequently the capacity of the low-pressure cylinders of the compound engine must be equal to the capacity of all the cylinders of the simple expansion engine.

Mechanical advantages of compound and triple expansion engines.—A great advantage of the stage expansion engine, so far as its mechanism is concerned, is the facility with which it allows high rates of expansion of steam to be carried out without bringing excessive stresses on the framing. As an example, if we consider the cases of two engines working at the same number of revolutions, one simple, the other compound, each supplied with steam of 60 lbs. initial pressure, and developing 2,100 I.H.P. with a total rate of expansion of 8 times, we shall find that whilst the maximum turning moment in the case of the compound engine is 960 inch-tons, it would be 1,250 inch-tons in the engine with simple expansion, or more than 30 per cent. greater, the mean moment and therefore the horse-power being the same in the two engines.

In consequence of the greater uniformity of twisting moment, the shafting and framing may be made lighter in the compound than in the simple engine, and much greater steadiness of motion may be obtained, and more efficient action of the propeller in the water expected. The great variations of pressure to which the shafts of simple engines are exposed when worked at high rates of expansion appear to produce the same effect on the material that vibration does, viz. to cause the structure to become crystalline. Several cases of broken shafts in engines of this class were attributed to the excessive intermittent stresses brought on them.

In the compound engine, although the steam is expanded 8 times when developing full power, it can be expanded still more when working at reduced powers, whereas in the non-compound engine, the steam being expanded in a single cylinder, it cannot be expanded much more than 8 times, whatever the reduction in the power may be. This results from the necessary mechanical arrangements, and is altogether independent of any loss of efficiency that would ensue from liquefaction, &c., when attempting to carry out a high rate of expansion in a single cylinder.

The superiority of the stage expansion engine is further demonstrated as the engine becomes worn. When the slides and pistons begin to leak, the loss in the simple engine is much greater than in the compound, in consequence of the greater difference of pressure in the former.

The steam leaking past the piston in the simple engine goes direct to the condenser without doing any useful work, whilst in the compound engine the steam leaking past the high-pressure piston does useful work in the low-pressure cylinder before passing to the condenser, and the amount of leakage in the low-pressure cylinder is reduced on account of the considerably smaller difference of pressure on the two sides of the piston in that cylinder.

Use of expansion valves and independent expansion fittings.—Since the cylinders of triple expansion and compound engines provide

in themselves for a considerable amount of expansion, special cut-off or expansion valves are now dispensed with, thus reducing the complexity and number of parts, as compared with the simple engine, in which expansion valves, suitable for early cut-off, are a necessity when high-pressure steam is used.

Many of the early compound engines, however, were fitted with expansion valves on the high-pressure cylinder, and some had expansion valves on the low-pressure cylinder also, in order to regulate the proportionate amount of work done by the two cylinders and equalise the stresses on the machinery. Without this valve, or some equivalent, at very low powers, as in warships on ordinary service, the work done in the low-pressure cylinder becomes very small. By setting the low-pressure expansion valve to an early cut-off, the pressure in the receiver, which forms the back pressure in the high-pressure cylinder, would be increased, so that the work done in that cylinder would be diminished and that in the low-pressure increased, and the power would consequently be more equally divided between the two cylinders.

Separate expansion valves are not now fitted to the cylinders, but to allow of adjustment in the points of cut-off, the reversing arms of the engines are now fitted with sliding blocks to enable the slide-valves to be linked up independently of the high-pressure valve, so as to vary the amount of expansion (see Chapter XV.).

Triple expansion engines generally.—The arguments which prove the superiority of the ordinary compound engine over the simple expansion engine when the working steam pressures were increased from 30 to 60 lbs. per square inch, also explain the superiority of the triple expansion engine over the compound engine for steam pressures above 120 lbs. per square inch. The principal gain in each case is the increased economy due to the greater amount of expansion conveniently obtained, and the reduction of the variation in temperature of the cylinders, which decreases the loss from liquefaction. Further, as previously mentioned, there is a more regular turning moment on the shafting, and a great reduction of maximum stresses on the engine and framework.

Some of the forms in which the triple expansion system has been carried out are illustrated in Figs. 185 to 195.

The arrangement shown in Fig. 185, has the high-pressure and intermediate pistons on the same rod, with the low-pressure acting on a separate crank at right angles to the other. Though convenient in some cases, this cannot be considered altogether satisfactory, as the stresses on the crank-pins would be very unequal.

In Fig. 186, each of the cylinders is fitted over a separate crank, the high, intermediate, and low-pressure cylinders being arranged in succession. This is the usual arrangement of triple expansion engines of moderate power both in the Royal Navy and mercantile marine. The cranks are arranged at equal angles with each other, although other arrangements have sometimes been fitted. The direction of revolution when going ahead may be either with the high pressure in advance of the intermediate, or the reverse.

Four-cylinder triple expansion engines.—For large powers, especially with quick-running engines, the low-pressure cylinder becomes so large as to require to be divided into two parts. although this generally

necessitates four cranks, and an increase in the length of the engine room. We thus get the four-cylinder triple expansion engine fitted in H.M.S. 'Powerful' and 'Terrible' of 25,000 I.H.P., also in all subsequent battleships and cruisers where the horse-power exceeds 10,000, as shown in Fig. 187.

Fig. 188 shows another form suitable for high powers, which can be fitted in cases where a considerable height is available for the machinery, as is often the case in the mercantile marine. This does not require a great length of engine room, and produces fairly uniform strains on the shafting, the high and intermediate-pressure cylinders being above the two low-pressure cylinders.

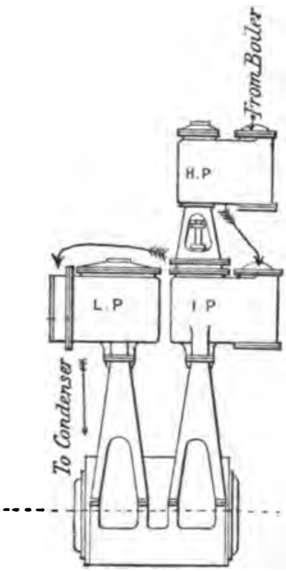


Fig. 185.

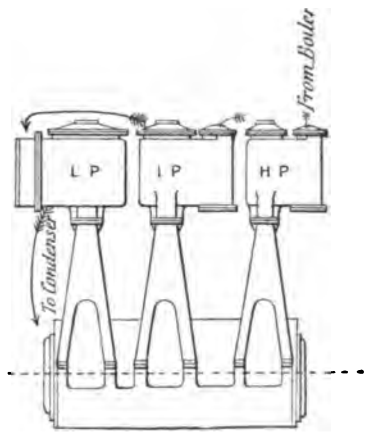


Fig. 186.

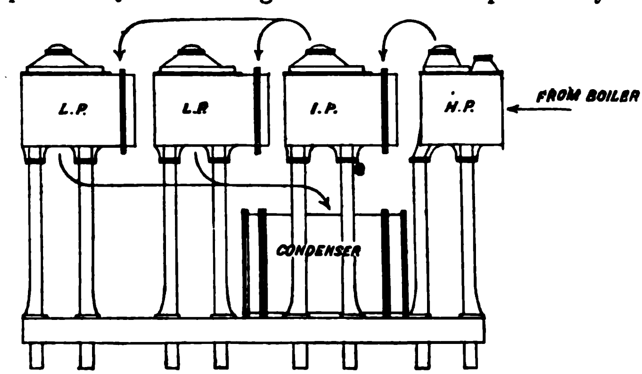


Fig. 187.

As previously pointed out, the relative positions of the cylinders do not affect the distribution of the steam, and are settled entirely by practical considerations. The four-cylinder triple expansion engine is

now the most usual type for large triple expansion engines, and may be taken as the standard type of modern marine engine now being fitted in the Royal Navy, and a common type for the high power vessels of the mercantile marine. The proportions and relative situation of the four cylinders and the arrangement of the slide casings and other details differ considerably.

Usual arrangement of triple expansion engines.—A vertical and horizontal section through the cylinders and slide-valves of the most usual type of large triple expansion engine is given in Figs. 189 to 192. In this example it will be seen that circular slide-valves are fitted for the high-pressure and intermediate cylinders, and flat valves for the low-pressure cylinders. In this respect engines by various makers differ from one another.

The arrangement as drawn represents that being fitted in the majority of new naval engines. In this service some few examples have flat slide-valves fitted on the high-pressure cylinder also, while in some of the earlier examples circular slide-valves have been fitted for all three cylinders. The Admiralty now specify that flat valves shall be fitted in the low-pressure cylinder and circular slides on the high pressure, the type fitted on the intermediate-pressure cylinder being also circular with the higher pressures of steam. Some circular slide-valves fitted were found difficult to keep steam-tight and allowed direct passage of steam to the exhaust side of the valve, and this is most objectionable in the low-pressure cylinder. Also, in this cylinder as the pressure forcing a flat valve against the cylinder face is not great there is no objection to its use.

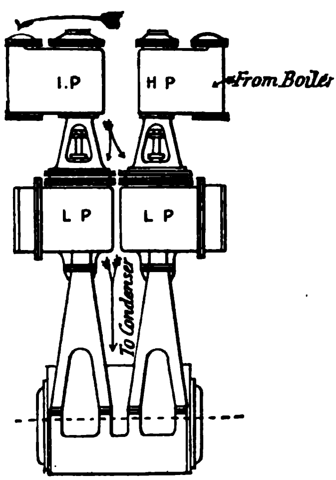


Fig. 188.

Ratio of cylinders.—In the mercantile marine the ratio of low-pressure to high-pressure cylinder-volumes with triple expansion engines generally varies from 6 at 140 lbs. steam pressure, to 7 at 160 lbs. pressure, and $7\frac{1}{2}$ at 180 lbs., and rather an earlier cut-off in the high-pressure cylinder is arranged for than is usual in the Navy. In the Navy smaller ratios of cylinders are fitted for the following reasons. In the first place the naval engine seldom works at full power, this being reserved for special occasions, while the mercantile vessel generally works at near full power. The greater part of the steaming in the Navy is performed at 9 or 10 knots speed requiring only a small fraction, say, on the average, for ships of high power about one-tenth the maximum.

It is also of great importance to keep the weight and space occupied by the machinery as small as possible, as very large powers are provided. To reduce the weight, therefore, the size of cylinders and amount of expansion allowed at full power is limited, as the economy at the highest power is not so important as that at lower powers. Limiting the size of the cylinders conduces to greater economy at low powers,

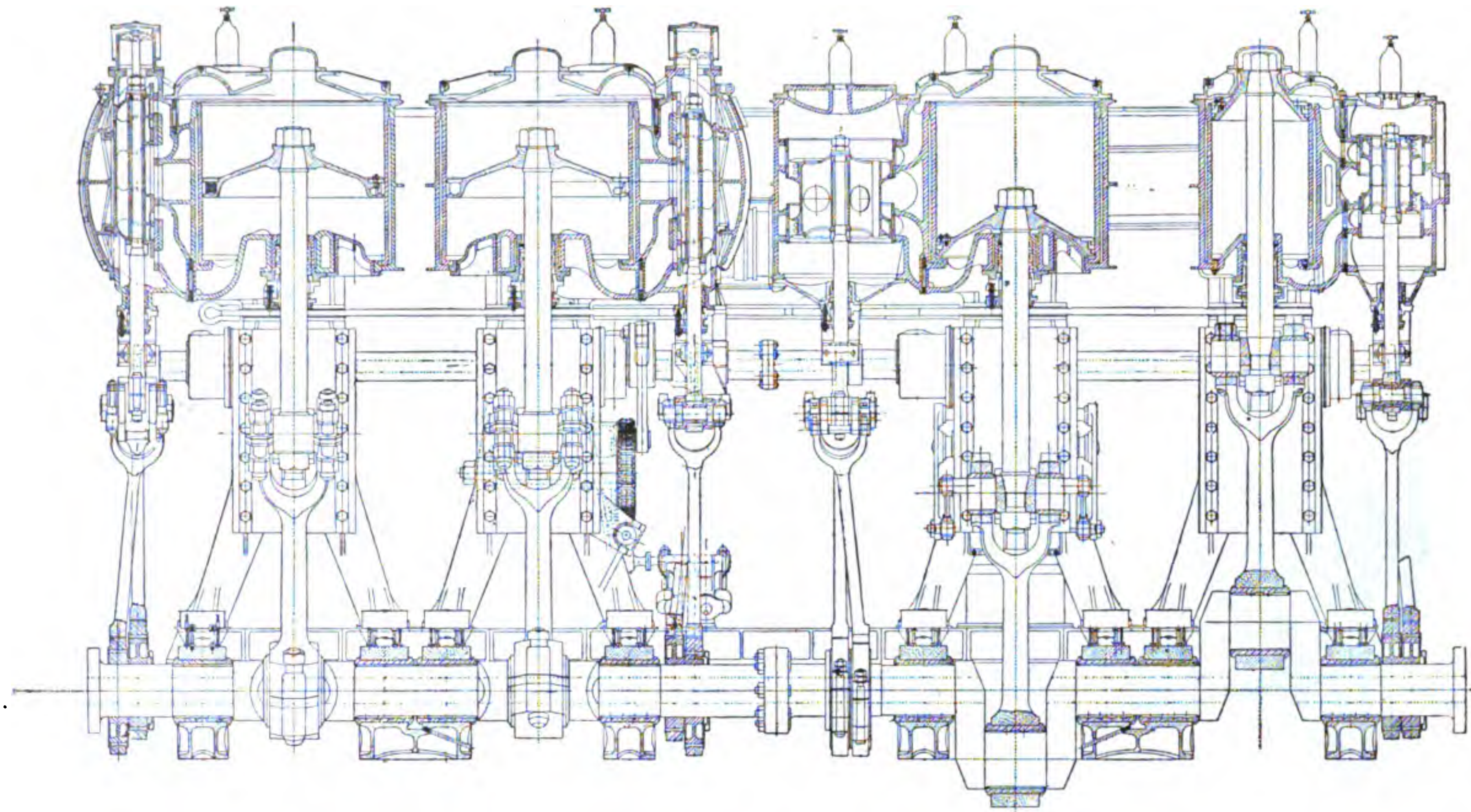


FIG. 189.

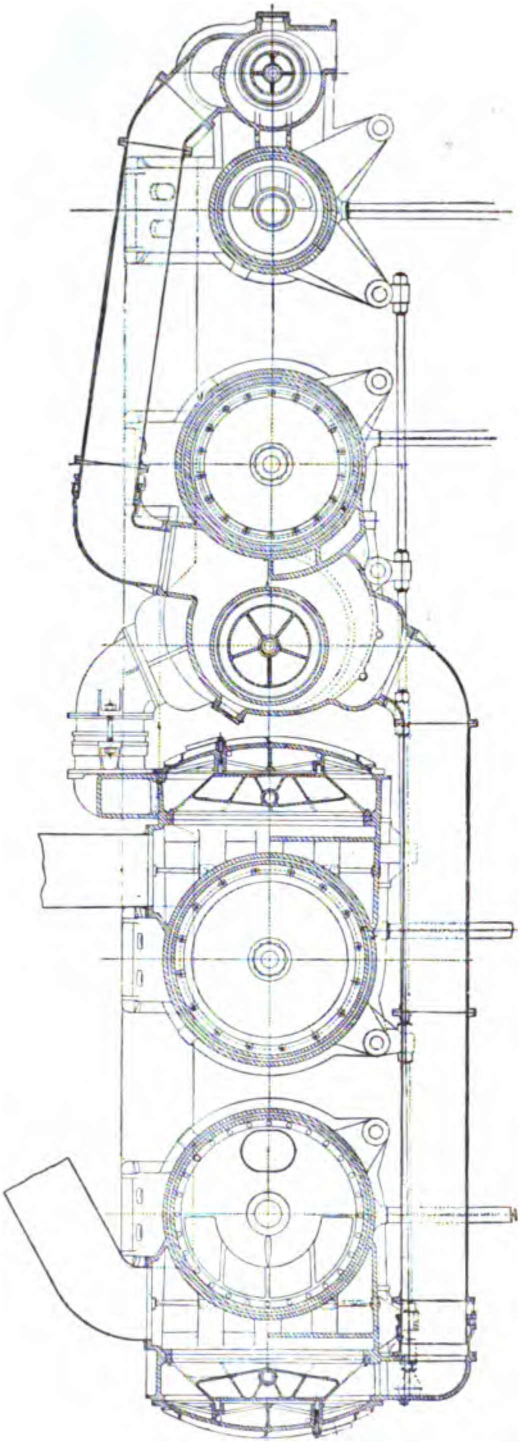


Fig. 190.

as with large cylinders and considerable expansion at full power the limit of economical expansion is very soon reached when the power is reduced, while with the smaller cylinders and less ratio of expansion at full power, there is a greater range for the utilisation of the maximum amount of expansion combined with an unreduced pressure of steam.

The ratios of cylinder - volumes adopted in the Navy for 155 lbs. steam pressure are usually 1 : $2\frac{1}{4}$: 4·84 to 5 for H.P.: I.P. : L.P. ; for 210 lbs. it is 1 : 2·4 : 5·7 ; while for 250 lbs. it is 1 : 2·6 : 7. In the mercantile marine it is about 1 : 2·7 : 7 at 160 lbs. pressure.

Arrangements of cylinders and cranks in four-crank triple expansion engines. — In arranging the cylinders and cranks of these engines the designer has a considerable range of choice, so that actual engines show a great variety in this

respect. The usual plan is to arrange the two after cranks opposite each other in the end view, the two forward being also opposite but at right angles to the after ones. This is indicated in Fig. 193 for the

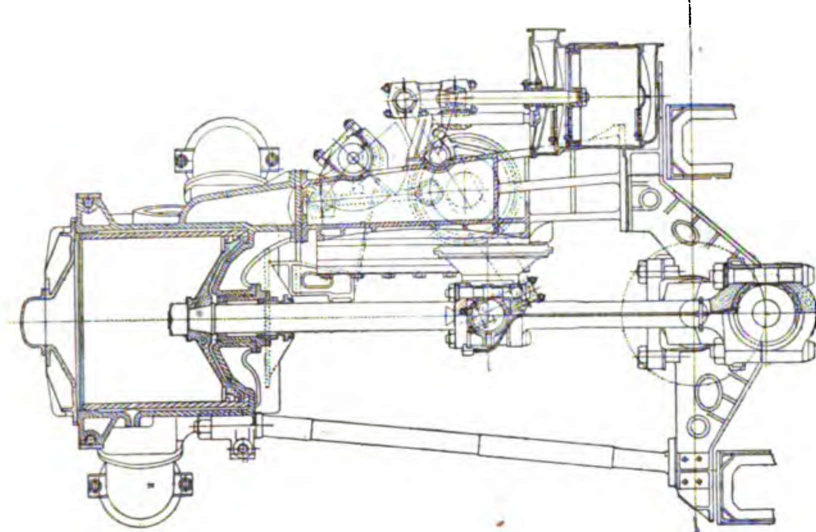


FIG. 192.

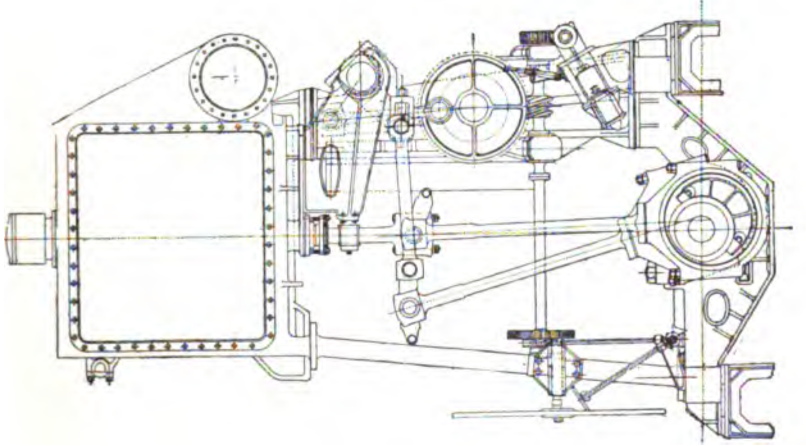


FIG. 191.

arrangement of cylinders shown in Fig. 190. Another plan is to place the two after cranks at right angles, with the forward ones opposite, as in Fig. 194. In a few examples the three-cylinder triple expansion arrangement has been repeated, the two low pressure cranks

being placed at the same point in the circle, and the high-pressure and intermediate at angles of 120° with them. The two first-mentioned plans were tried in the 'Powerful' and 'Terrible' respectively. The second plan gives a more uniform turning moment on the crank shaft, and favours starting the engines, while the first plan reduces the magnitude of the reciprocrating forces which cause vibration of the vessel.

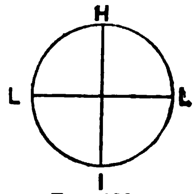


FIG. 193.

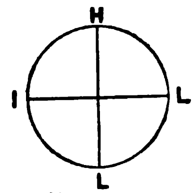


FIG. 194.

In the 'Terrible' the vibration was so considerable at certain speeds that the cranks were altered and a series of experiments carried out with different settings of cranks. The 'Powerful' arrangement, Fig. 193, was found much superior to that at first fitted in 'Terrible,' Fig. 194, while with two other arrangements tried, viz. Figs. 194*a* and 194*b*, the vibration was much further reduced. Fig. 194*a* was found to be difficult to start in certain positions, so that Fig. 194*b* was adopted with satisfactory results, and represents the setting of cranks now in the vessel.

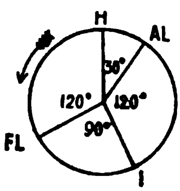


FIG. 194*a*.

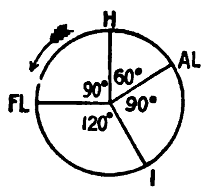


FIG. 194*b*.

In the arrangement of cylinders and slide-valves in the fore and aft direction, also, there are many varieties. Fig. 187 shows the

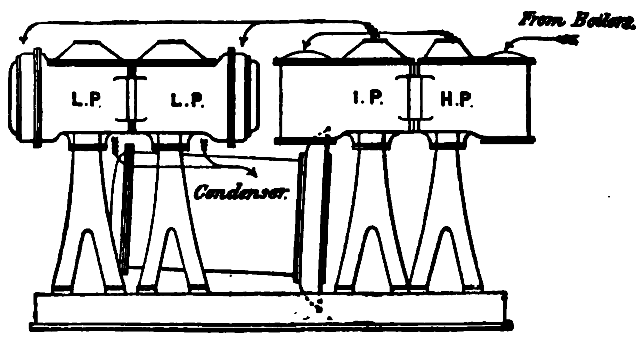


FIG. 195.

arrangement of H.M.S. 'Powerful' and others, while Fig. 195 shows a later arrangement in many large recent vessels of the Royal Navy, adopted when the importance of reducing vibration forces was realised. The two end cylinders are brought as near together as possible, with valve gears outside, while the forward pair of engines is served similarly. As each end pair acts on cranks practically opposite one another, the rocking moments tending to set up vibration are thus reduced. In many recent cases the two low-pressure cylinders are placed forward and aft

respectively, the valve gears being outside; the centres of each end

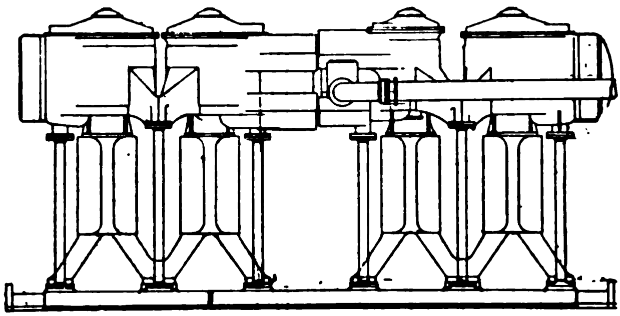


FIG. 195a.

pair of cylinders are thus brought rather closer together than with the preceding plan, while the cranks are similarly arranged, *i.e.*, the H.P. and forward L.P. cranks would be opposite or nearly so, while the I.P. and aft L.P. would be practically opposite but at right angles to the forward pair. These two arrangements are probably on the whole the best for four-cylinder triple expansion engines. In some designs for new American battleships the two L.P. cylinders are placed together in the middle, the H.P. and I.P. cylinders being at the forward and after ends respectively. There are also other arrangements, but the remainder do not possess any importance. In many vessels with divided L.P. cylinders the L.P. reciprocating parts are made much lighter than those of the H.P. and I.P. to correspond with the smaller amount of work done.

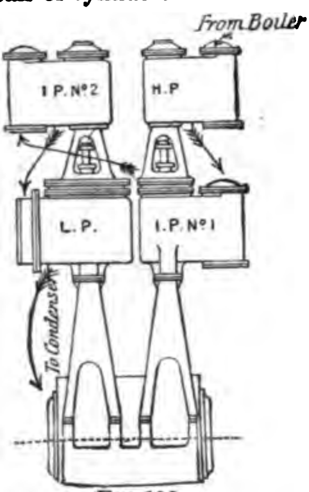


FIG 196.

Quadruple expansion engines.—In the mercantile marine a considerable number of quadruple expansion engines have been fitted, by

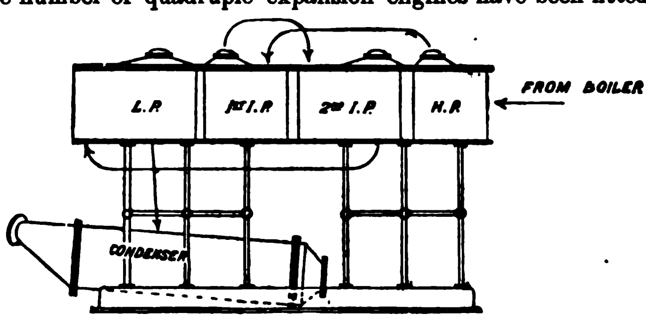


FIG. 197.

which the expansion of the steam is split up into an additional stage and

further economy obtained. The average steam pressure in engines recently built is 215 lbs. per square inch, while the ratio of cylinder-

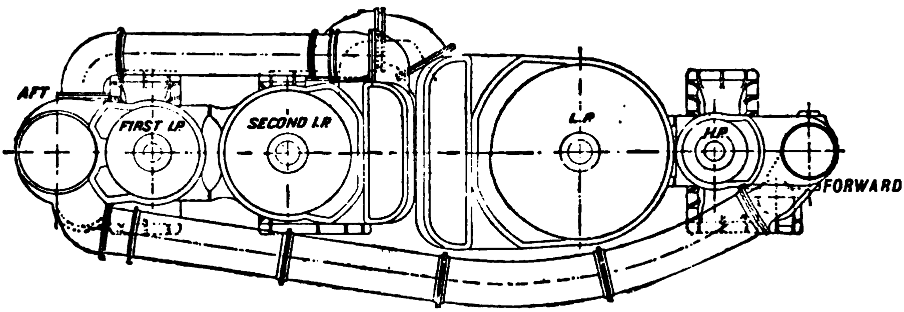


FIG. 198.

volumes varies. In one case the ratios are 1 : $2\frac{1}{8}$: $4\frac{1}{4}$: $8\frac{1}{2}$, and in several others the average is 1 : 2 : $4\frac{1}{3}$: $10\frac{1}{4}$ for the high-pressure, first intermediate, second intermediate, and low-pressure respectively, the cut-off of steam being about 70 per cent. in the high-pressure cylinder at maximum power.

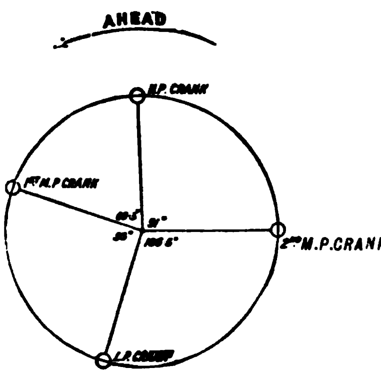


FIG. 199.

Fig. 196 shows an arrangement sometimes fitted where fore and aft space is important, and there is sufficient head room. Fig. 197 shows that of No. 90 torpedo boat, the only vessel in the Royal Navy fitted with quadruple expansion engines. The relative arrangement of cylinders, slide-valves, and cranks in quadruple - expansion engines varies still more than in the four-crank triple expansion engine, and in this case, again, the low-pressure cylinder has been sometimes divided into two parts, and five cylinders and cranks fitted, as in the s.s. 'Inchmona,' a vessel working at 250 lbs. pressure.

Fig. 198 shows an arrangement of cylinders, and Fig. 199 the angles of cranks in a large mail steamer, designed with the special object of reducing the forces causing vibration. The H.P. cylinder is forward, and the first intermediate aft, the others following in sequence. The weights of the smaller pistons are usually increased beyond the ordinary amount, while the cranks are shifted from the normal right-angled positions by amounts given by calculation, to assist this object. This arrangement is known as the Yarrow, Schlick, and Tweedy system, and is also applied to four-cylinder triple expansion engines.

CHAPTER XIX.

DETAILS OF CYLINDERS AND ENGINE-ROOM FITTINGS IN CONNECTION.

WE now describe the details of the cylinders and fittings.

Figs. 200 to 203 show sections of two cylinders, with pistons, slide-valves, &c., complete, and from these sketches the general form and arrangement may be understood.

The structure of a large cylinder is composed of three separate principal parts, viz. (*a*) the cylinder casting, or shell containing in one casting the outer framework of the barrel, the cylinder bottom, and on one side the passages or ports through which the steam is admitted to and discharged from the cylinder; (*b*) the cylinder liner, which is bored to a true cylinder and forms the steamtight surface on which the piston actually works; (*c*) the cylinder cover, which closes the open end of the cylinder and completes the steamtightness at this part.

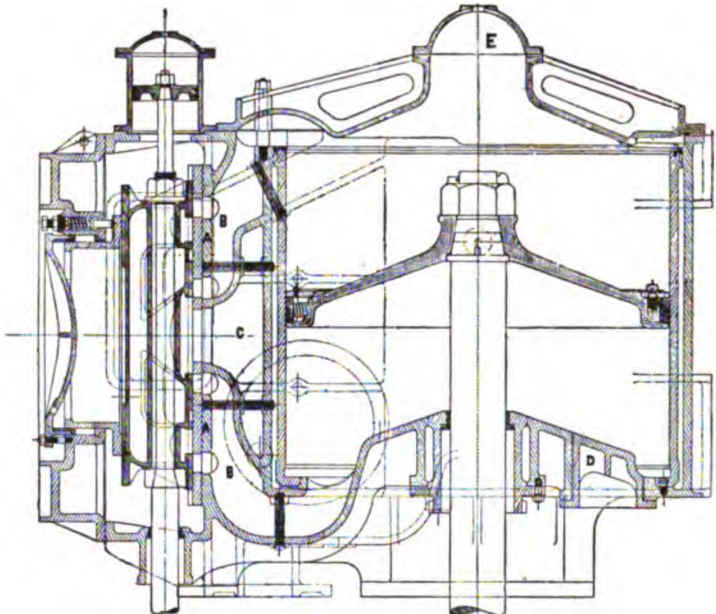
The cylinder shell.—From an examination of the sketches it will be seen that this casting is of a complicated nature, and to insure a sound and satisfactory casting, a brand of cast-iron is used which is suitable for running freely in the moulds prepared. This brand of cast-iron, although very suitable for this purpose, is too soft to be used for the rubbing surfaces, so that these latter are separate, and are constructed of a harder variety of cast-iron, and owing to the simple form of these rubbing surfaces, no difficulty is experienced in satisfactorily casting them in the harder metal. These rubbing surfaces are the cylinder liner on which the piston works, and the cylinder slide face on which the slide-valve works. Owing to its complicated form it is not possible to make the cylinder with ports in steel.

The cylinder slide face A should be arranged as near as possible to the cylinder barrel, consistent with obtaining sufficient area for the inlet of steam through the steam ports B, and for free exhaust through the exhaust port C.

On the bottom of the cylinder are cast the feet which are secured to the columns on which the cylinder is supported.

The lower or crank end of the cylinder being usually cast with it, forms a part of the cylinder itself, and in the centre of this end there is a hole, which allows the boring bar to be passed through, for the purpose of boring out the internal surface. This hole is afterwards closed by a door or plug, which is jointed so as to be steamtight, and which carries the piston-rod stuffing-box and gland. In large cylinders, where there is sufficient room between the piston-rod stuffing-box and the cylinder barrel, a manhole and cover, D, are fitted to the bottom of

Fig. 200.



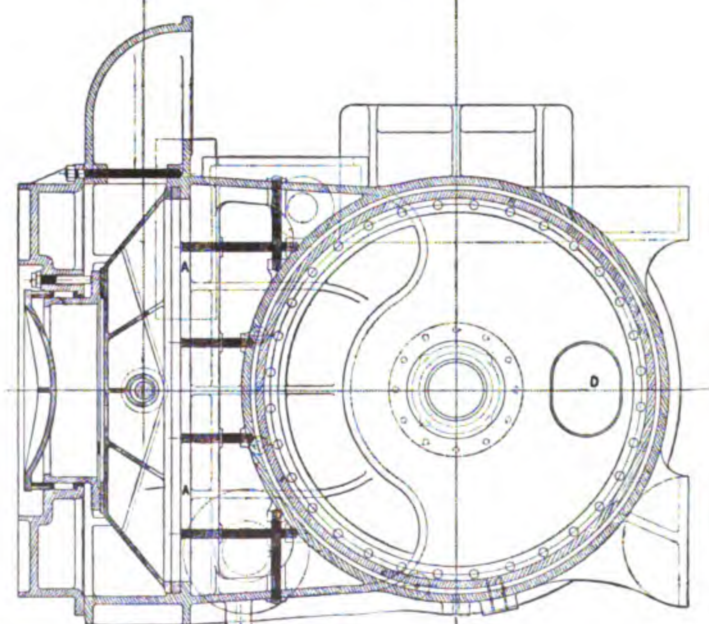


Fig. 201.

the cylinder, to enable examination to be made without removing the cylinder cover and piston.

Fig. 202.

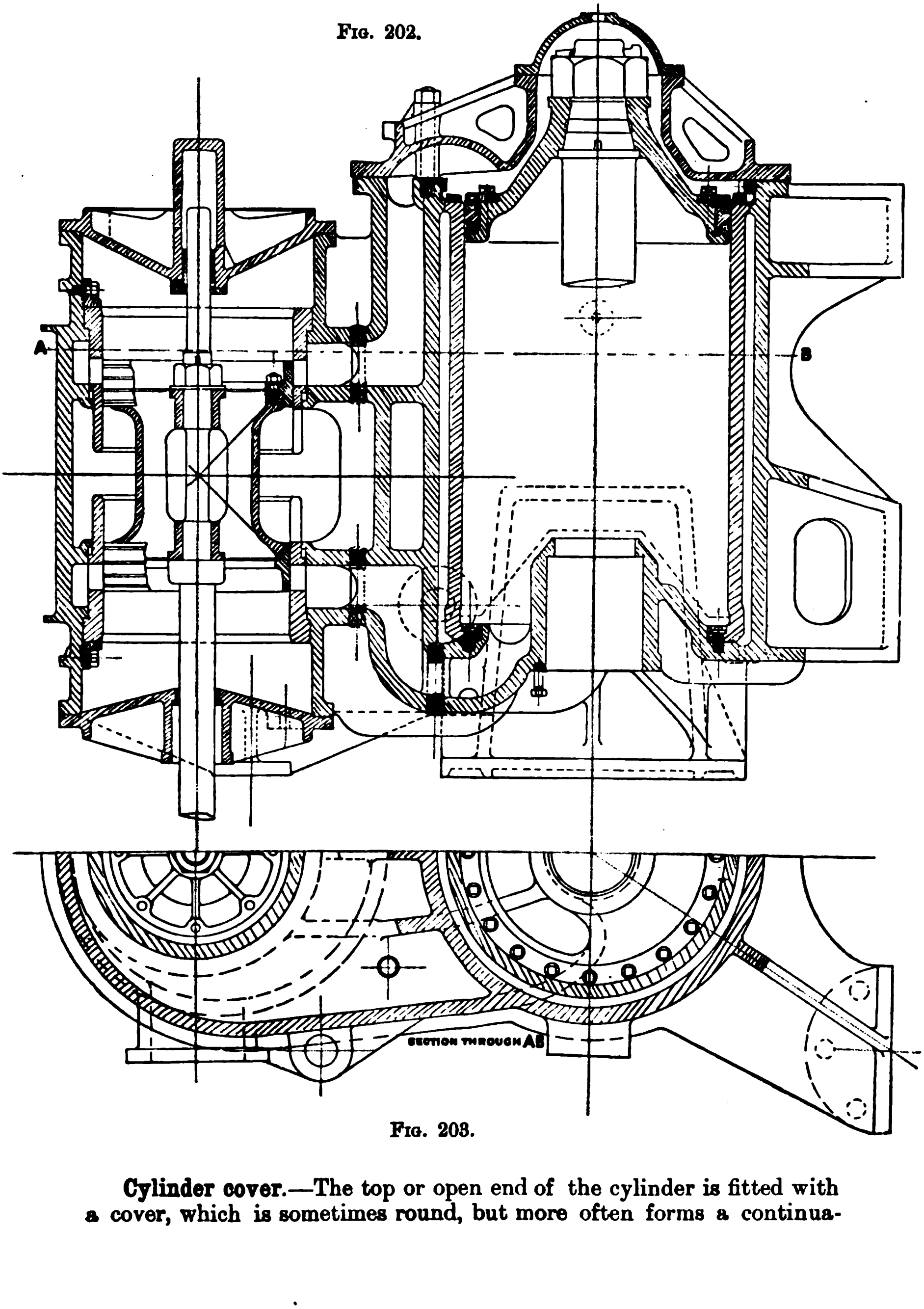


Fig. 203.

Cylinder cover.—The top or open end of the cylinder is fitted with a cover, which is sometimes round, but more often forms a continua-

tion of the upper steam port, as shown in Figs. 200 and 202. This cover is also fitted with a manhole, E, in the centre in order to avoid repeatedly breaking the large joint between the cover and cylinder for examination purposes. This cylinder cover is generally made of cast-steel and formed of a single wall of metal stiffened by deep radial ribs.

The cylinder end and cover were at one time cast hollow, the space between the two plates being kept full of steam when the engine is at work, and forming part of the steam-jacket, but this is generally not now so fitted. The general construction of the cylinder end is now as shown in the sketch, in which the end has only one wall of metal, but is stiffened by numerous radial ribs.

Where piston-valves are used the construction of the steam ports is different, and this is shown in Figs. 202 and 203, which gives details on a larger scale of the H.P. cylinder of the same engine, the low-pressure of which is shown in Figs. 200 and 201. In both these examples the cylinder ports are supported and strengthened by numerous ribs and screwed stays.

Cylinder liner.—The working barrel is secured by a flange at the bottom end, fitted with bolts which are generally recessed into the flange. In the Royal Navy, in consequence of some cases having occurred in which these bolts have slacked back, they are secured from turning. Fig. 204 shows the head of the bolt slightly hammered out into a groove formed in the recess for this purpose. The cylinder cover end is left free to allow for expansion. The joint is kept steamtight, by fitting either a small stuffing-box as shown, packed with asbestos or other material; or a copper ring, of the section shown in Fig. 205, which also allows the necessary expansion. This latter plan is considered preferable, a permanent and lasting joint being the result. The space between the liner and cylinder is usually from $\frac{3}{4}$ to $1\frac{1}{4}$ inch in depth, and is generally kept filled with steam, thus forming the steam-jacket, and also permitting of the gradual warming of the cylinders when raising steam prior to starting.

There are many advantages resulting from this method of construction. It very much reduces the complexity of the casting for a jacketed cylinder. In many cases in which cylinders have been made with the inner and outer barrels in one casting, the unequal contraction of the metal in cooling has caused excessive strains on parts of the material, which developed into cracks by the working of the engines and gave much trouble and anxiety.

With the separate liner or barrel, it is also more easy to insure that the working surface should consist of hard and sound material, so that the friction may be decreased and the durability of the cylinder increased, and when the working surfaces of the cylinder become much worn, the liner may be renewed at a comparatively small cost. Recent practice has been to fit hard close-grained cast iron for these working barrels, but the high pressure is often made of forged steel which is sometimes hydraulically compressed.

Cylinder face.—The face, on which the slide-valve works, is now generally cast separate from the cylinder, as shown in Fig. 200, and secured by a number of gun-metal or naval brass cheese-headed screws, with heads recessed to some depth below the surface of

the false face. These recesses act as small oil-cups or reservoirs, and assist the lubrication. The advantage of this arrangement is that good sound hard metal can be insured for the working faces, and in case of wear, the face can be easily renewed. The faces are made of hard close-grained cast-iron. Phosphor-bronze was tried for these faces some years ago, but it was found to be inferior to good cast-iron.

Clothing or 'lagging' of cylinders.—In addition to the steam-jacket, which is fitted to nearly all large modern engines, it is necessary that the outside surfaces of cylinders and slide-jackets should be carefully covered or clothed with non-conducting material to prevent radiation. Where the temperature is great the non-conducting material is one that is also incombustible, to avoid the charring which would otherwise occur. The high-pressure and intermediate cylinders are generally lagged in this manner, but in some cases a thin foundation of incombustible material, such as sheet asbestos, is fitted next to the hot surface, and the necessary thickness made up with other non-conducting material, not incombustible. The clothing material is usually kept in place by an outer covering of wood, or sheet steel or iron, the sheet material being preferred, as it lasts longer and can be more readily taken off without damage when necessary for any purpose.

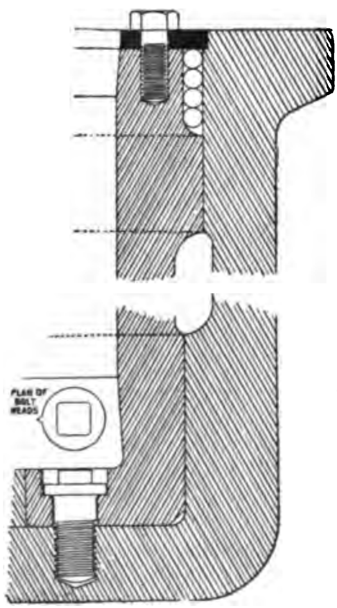


FIG. 204.

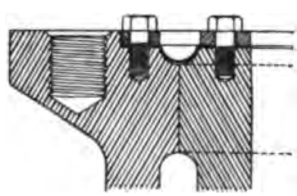


FIG. 205.

Pistons.—The piston is the agent by which the energy exerted by the steam is transmitted to the mechanism of the engine. It was for many years made of cast-iron of hollow form, and stiffened by internal ribs, which was a strong construction. This old form is shown in Fig. 184, but it is now generally made of cast-steel, and its usual construction is as shown in section in Figs. 206 and 207, which give details of two pistons of a triple expansion engine. By this use of cast-steel for pistons a saving of weight of about 40 per cent., as compared with cast-iron, has been effected, for pistons of cast-steel, on account of superior strength, are made of a single thickness, and of conical or dished form to give stiffness.

The piston consists of three principal parts, viz. the piston body,

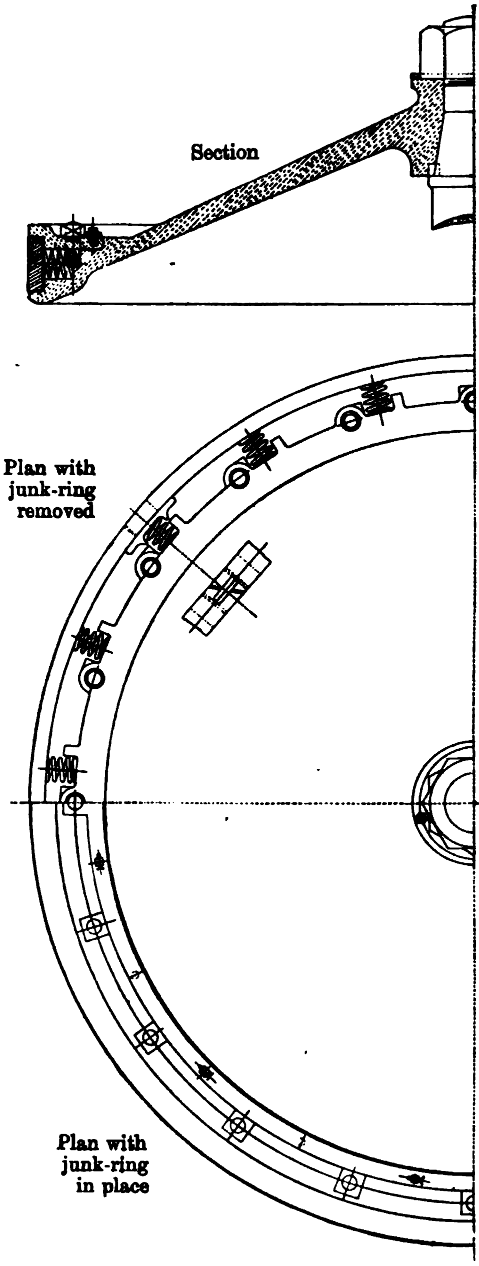


Fig. 206.

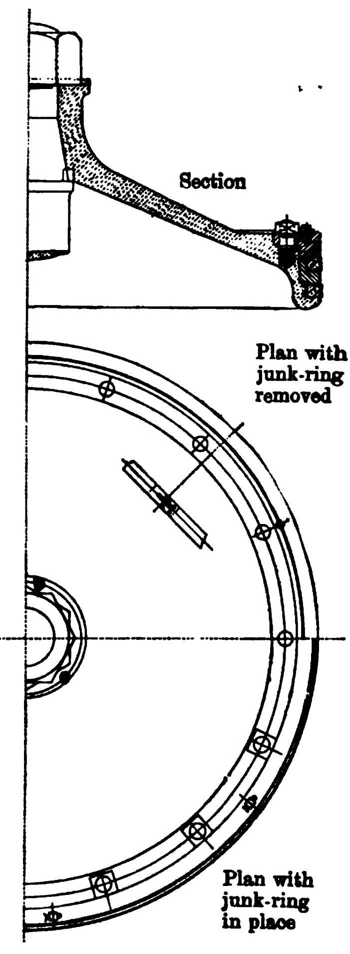


Fig. 207.

the packing-ring which works on the cylinder barrel, and the junk-ring which secures the packing-ring in position.

The piston should work steamtight in the cylinder in order to prevent leakage from one side to the other, which causes waste of steam and needlessly increases the back pressure. It should also be quite rigid in resisting any tendency to deformation due to the steam pressure acting alternately on its faces, and it should also move in the cylinder with as little friction and wear of rubbing surfaces as possible. Its attachment to the rod should be of the firmest possible description.

Packing-ring.—In nearly all old marine engines, and also in the low-pressure pistons of stage-expansion engines, the steamtightness is accomplished by means of a metallic *packing-ring* of considerable depth, sometimes called the *spring-ring*, which is kept pressed against the surface of the cylinder by the action of steel springs. Some makers use springs similar in form to coach-springs, as in Fig. 208; others use complete circular springs of various forms to press the packing-ring against the cylinder; and in other cases spiral springs are fitted in recesses in the body of the piston. This latter plan is shown in Fig. 206, and is now preferred and specified for engines of the Royal Navy, as the pressure exerted by the springs against the packing-ring, which forces the latter against the cylinder barrel, can be regulated and adjusted as required, and is always known. With coach-springs it is always a very variable and uncertain amount. The spiral springs are compressed so as to exert a pressure of about 2 lbs. per square inch of the bearing surface of the packing-ring.

In order to allow the ring to naturally tend to spring tightly against the cylinder or liner, it is made of slightly larger diameter than the latter; a piece is then cut obliquely out of the circumference, and the ring closed to fit the barrel, so that the effort of the ring to regain its original diameter helps to keep it tight against the cylinder.

Tongue-piece.—To prevent the passage of steam from one side of the packing-ring to the other through the oblique cut, shown at *a b* (Fig. 209), a groove, *c d*, is cut in the adjoining edge, and a gunmetal plate, A, is fitted behind the joint, with a tongue-piece to fit tightly in this groove. The plate is secured to one end of the metallic packing-ring rigidly, and to the other by bolts in elongated holes, so that, as wear occurs the ring expands, and still keeps steamtight on the cylinder surface. This construction applies principally to broad packing rings, but narrow rings are sometimes fitted similarly, as shown in Fig. 207.

In many cases, in order to prevent excessive pressure on the cylinder, should steam obtain access to the back of the packing-ring, the tongue piece is so fitted as to prevent the ring from increasing in diameter beyond a certain amount; but recent experience in the Royal Navy is in favour of forming small projections on the packing-rings, as shown at B in Fig. 208*a*, fitting in corresponding recesses, which prevent the rings from increasing in diameter more than a small amount. Fig. 208*a* shows a common construction for the rings of H.P. pistons.

Junk-ring.—One edge of the packing-ring is in contact with a rim on the piston, and it is kept in its place by an annular plate called the '*junk-ring*,' firmly bolted to the other side of the piston (see Figs. 206 and 208). The edges of the packing-ring, and the faces of the piston and junk-ring in contact, are carefully fitted together so that the

joints may be steamtight. The origin of the term 'junk-ring' may be traced to the time before metallic packing was introduced, when the pistons were packed with hemp gasket or junk packing, and the office of the junk-ring was to keep this packing in place, and press it against the cylinder, to keep the piston steamtight.

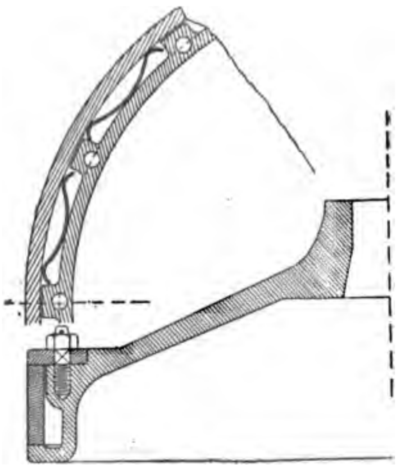


FIG. 208.

The figures last mentioned show the arrangement of the springs, &c., and the method of attaching the junk-ring to the piston. When the pistons were of cast-iron, brass nuts were let into the body of the piston, into which the junk-ring bolts were screwed, as the junk-rings have to be frequently taken off for the examination of the springs, and if the bolts were screwed into the body of a cast-iron piston they would soon become slack. With the separate brass nuts defects are less likely to occur, and can be readily made good by the fitting of new nuts and bolts. With steel pistons these recessed nuts are not usually fitted, but either steel studs are fitted in the piston with gunmetal nuts, or the junk-ring bolts are made of naval brass or some similar material, so that they will not rust fast in the piston body, and when worn can be readily renewed. For horizontal engines the springs are not continued all round the piston, but solid blocks, termed 'cod pieces,' are substituted for them at the bottom, for about one-fourth of the circumference, to assist in supporting the weight of the piston, while in some vertical engines similar blocks are fitted, instead of springs, on each side of the piston to resist the forces caused by the rolling of the vessel, which tend to cause the piston to press on one side of the cylinder barrel.

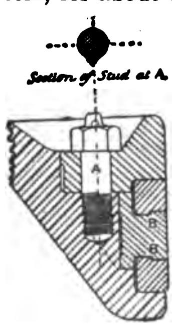


FIG. 208a.

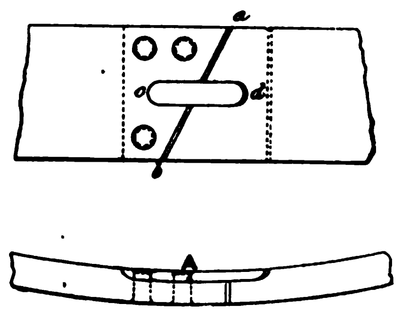


FIG. 209.

Special piston packings.—Large numbers of patent piston packings have been devised, many of which have split packing-rings, which aim

at causing the springs not only to press against the cylinder barrel, but also to press against the faces of the junk-ring and piston, and prevent steam passing to the back of the spring ring. One of them (Lockwood and Carlisle's) is shown in Fig. 210. It consists of a split packing-ring, containing a compound spring constructed as shown. The helical parts of this compound spring press the spring ring against the cylinder, while the remaining portions press the two halves of the packing-ring against the junk-ring and piston flange respectively, so as to keep those joints steamtight. The junk-ring compresses the springs about $\frac{3}{16}$ inch.

With steam of very high pressure, should the junk-ring be badly

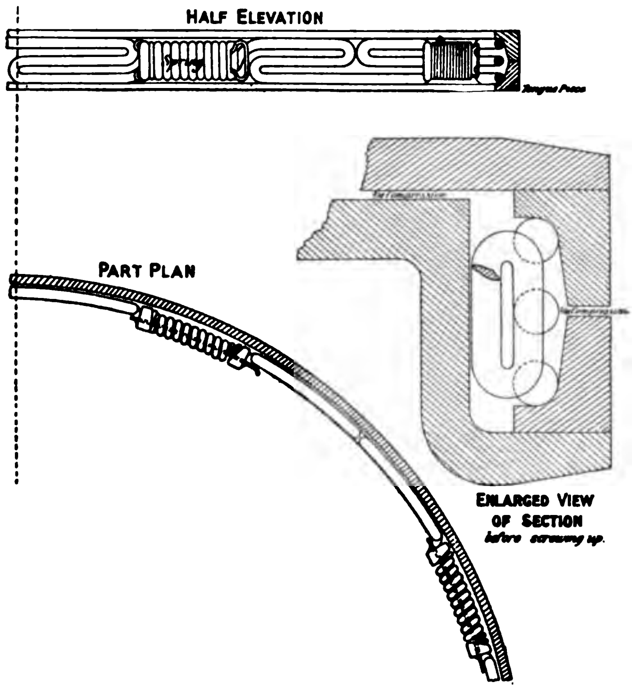


FIG 210.

fitted or worn at the edges, so that steam is allowed to pass to the back of the packing-ring, a great pressure acts, forcing it against the cylinder barrel, causing excessive friction and wear. Careful attention should therefore be paid to this part of the engine. With high-pressure steam the packing-rings of the high-pressure cylinder, instead of being of the broad single-ring type, are generally small square rings of cast-iron or special bronze without springs behind them, as in Figs. 207 and 208*a*. These large rings cannot be sprung into position, so that a carrier frame and junk-ring are necessary as before.

Guard-ring.—To prevent the possibility of any of the junk-ring bolts or nuts slacking back, a guard-ring is generally fitted to the heads of the bolts or nuts to prevent their turning after they have been

screwed up tight. This guard-ring is secured by square-necked studs, with nuts and split pins. Sometimes, however, the nuts of junk-ring studs are prevented from slacking back by stout split pins, the studs being square-necked. The former method is preferable as being less liable to be left out of place.

Ramsbottom rings.—In many small pistons, such as those for auxiliary engines, &c., two or more small rings, cut at one part of the circumference, are used instead of a single packing-ring. In this case the piston is solid, and neither junk-ring nor springs are required. Grooves are cut in the circumference of the piston into which the rings are sprung, the joints not being placed in the same line. These are known as Ramsbottom rings. The piston of the steam cylinder in Fig. 362 is fitted in this way.

Piston-rod.—The piston-rod is made of wrought-steel, and, except in the few special cases where it is secured by a flange, passes through the piston and is secured by a nut on the opposite side. The part in the piston is usually coned to allow it to be drawn up tightly, and fitted with a stop to prevent the piston turning. It is, however, sometimes made parallel, with a collar on the rod, to take the thrust of the piston. To some steel pistons the piston-rod is attached by means of a flange, as shown in Fig. 211.

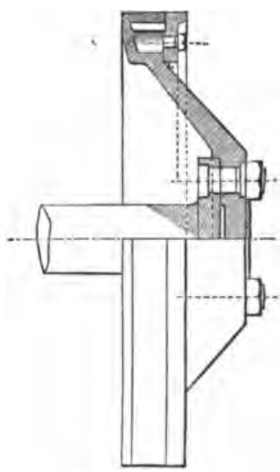


Fig. 211.

Sketches of piston-rods of different kinds are given in Figs. 212 to 215. The attachment of the piston to the rod should be very secure, and the rod should be very carefully fitted to the hole in the piston. A substantial pin or cotter is sometimes fitted through the end of the rod to prevent the nut unscrewing, and in this case the pin or cotter should be recessed into the nut as in Fig. 212. The most usual method is, however, to fit a plate around the nut, secured to the piston by studs with square necks, and having nuts secured by split pins. This plate is shown in Figs. 214 and 215. Considerable racking strains come on this part, especially if water should at any time accumulate in the cylinder, when a considerable bending moment acts, tending to bend the rod near the lower surface of the piston. For this reason it is undesirable that any diminution of section should be permitted at this part, and any collar desired should therefore preferably be obtained by an enlarged diameter.

Either a substantial collar or a sufficiently steep cone should be fitted, and in some cases both are found. In the latter case, sometimes both collar and cone are carefully fitted to bear on the piston together, or a small space is left between the collar and the piston to enable the former to come into operation should there be any yielding of the cone in the piston when at work. The removal of the piston-rod from the piston is often very difficult, and to facilitate this operation, when required, special fittings are generally provided for forcing it out. Two

plans of effecting this are shown by dotted lines in Figs. 213 and 215, which show the portable fittings which are applied after the piston rod nut has been removed. Fig. 213 is an example from a torpedo boat destroyer.

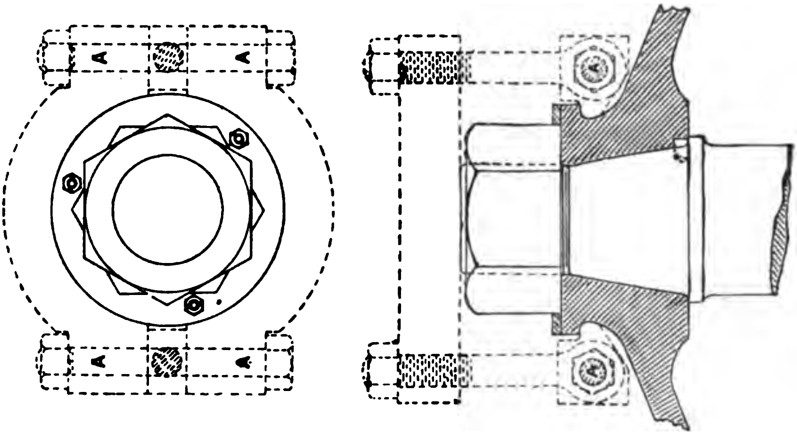


FIG. 215.

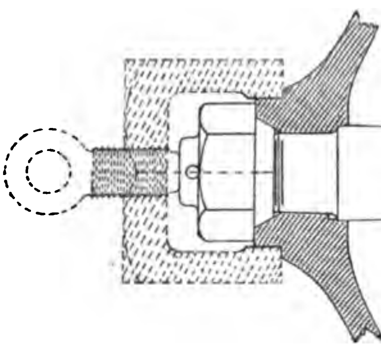


FIG. 213

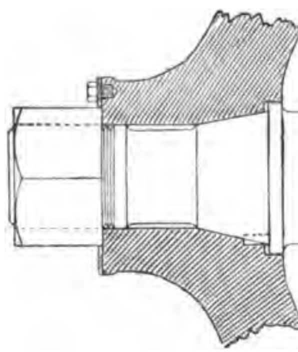


FIG. 214.

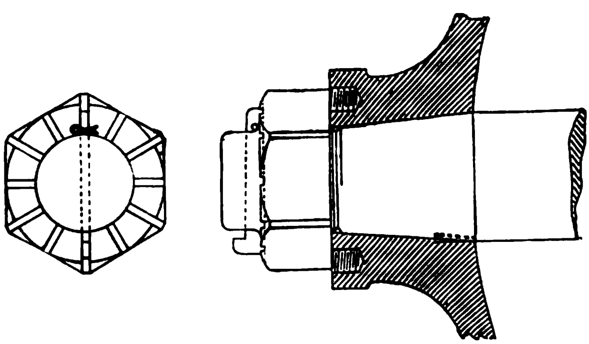


FIG. 212.

Supporting the weight of horizontal pistons.—Large horizontal engines are now obsolete, but when they were used means were pro-

vided to support the weight of the pistons, to prevent the wearing of the cylinder and metallic packing-ring at the bottom.

This was effected by a tail-rod or trunk at the cylinder-cover end of the piston, continued through a stuffing-box in the cover, with a block fitted on the end of it working on a suitable guide, so that the only pressure between the working surfaces of the piston and cylinder might be that due to the springs of the metallic packing-ring.

With vertical engines tail-rods fitted as continuations of the piston-rods, and working through stuffing-boxes in the cylinder covers, are often fitted in the mercantile marine, but in the Navy experience has indicated that they are unnecessary and may be objectionable, and they are not now fitted.

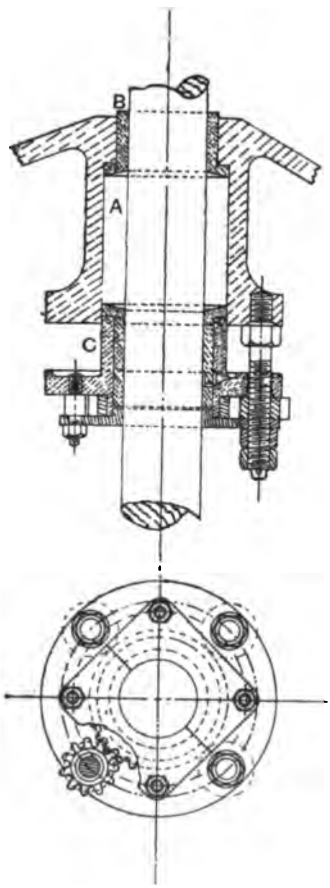


FIG. 216.

Stuffing-boxes.—The holes in the ends of the cylinder through which the piston-rods pass are fitted with stuffing-boxes, which permit the reciprocating motion of the rods to take place and prevent the escape of steam or admission of air to the inside of the cylinder. Stuffing-boxes are necessary in all cases in which a rod that must be free to move is brought through the end of any chamber that has to be kept steam, air, or water tight. This includes all piston-rods and pump-rods passing through the ends of steam or water cylinders, the slide and other valve-rods, the stern shaft where it passes out of the ship, and all similar fittings.

A general arrangement of a stuffing-box for ordinary purposes is shown in Fig. 216. It consists of an annular space, A, around the rod, which space is filled with material of an elastic nature, generally known by the name of *packing*. The inner, or cylinder end of the stuffing-box is fitted with a brass bush, B, tightly fitting the cylinder, but sufficiently large in its bore to allow the rod to move freely to and fro. This is termed a *neck bush* and its use is to provide a surface to retain the packing when the latter is pressed against it. The packing is pressed against the rod by a movable bush, C, called a *gland*, which has flange projections to enable studs and nuts to force the cylindrical part against the packing and press it against the rod, so as to keep it steamtight without unduly increasing the friction. The gland is either of brass, or of cast iron with a brass lining adjacent to the rod, and both gland and neck bush have conical ends, so as to assist in pressing the packing against the rod.

For the packing, gasket made by interweaving strands of hemp or cotton to form a rope may be used for pump-rods, stern glands, &c., but elastic core packing, which consists of a core of indiarubber, round which canvas or asbestos is tightly coiled, is sometimes used for such purposes. For steam rods asbestos fibre is generally used, except when the steam pressure and temperature are high in which case it is destroyed too rapidly and also tends to wear away the rod.

Defects of elastic or fibrous packing glands for high pressures.—Where the rod is kept steamtight in the preceding manner, if the steam pressure is considerable, the pressure exerted on the rod by screwing up the gland becomes very high, with the result that the friction is so great that the rod becomes heated and burns or chars any fibrous material in the stuffing box, and frequently rods so heated become bent. Even when the pressure is not so high as to cause this, or the material resists this charring action, there is a gradual wearing away of the rod and reduction of its diameter which causes it to be still more difficult to maintain steam tightness, while the friction causes an appreciable diminution of power.

Metallic packing for stuffing-boxes.—For high steam pressures, therefore, and especially when the rods are large, semi-metallic or entirely metallic packings are now fitted, and are essential for continued efficiency. There are many varieties, and a type which has been largely fitted is shown in Figs. 217 and 218. It consists of alternate rings of white metal A, and gunmetal B, with wedge-shaped sections, carefully fitted and scraped together. The white metal is in contact with the moving rod. These metal parts bear the contact with, and impact of the steam, without being soon destroyed, as soft packings are in such positions. The metallic packing thus fitted was not, however, absolutely tight and it was supplemented by a few turns of soft packing placed below the metallic part, the former is then removed from the destructive action of the high-pressure steam, and stops any small leakage of steam which passes the metallic packing.

For large rods the soft packing was fitted in entirely separate stuffing-boxes, as shown in Fig. 217, while for smaller rods it was fitted either in this manner or with the soft packing bearing on the metallic packing, as shown in Fig. 218. Many piston rods in the Royal Navy, and many slide-rods also, are fitted as in Fig. 217, while many small rods are fitted as in Fig. 218.

In the type shown in Fig. 217, where there is no means of adjustment for the metallic packing when under way, springs are fitted as shown, so that any slackness developed by wear in working may be taken up, so as to prevent movements of the packing in its gland.

Entirely metallic floating packing.—It will be noticed that the type of packing last described, while largely metallic, still contains some of the defects inherent in the use of soft packings screwed up by nuts. It also does not allow for the rod working slightly out of its proper line, as sometimes occurs, or for vibrations or tremors of the rod. These force the soft packing on one side and cause leakage of steam which, if prevented by additional screwing up, causes extra friction. The type of metallic packing illustrated in Fig. 218A overcomes these difficulties in a satisfactory manner. It is entirely metallic, self-adjusting, and works well, even when the rod is slightly

out of line, and experience with it has been very satisfactory both in the Royal Navy and Mercantile Marine. It is known as the 'United States' metallic packing.

There is more than one variety, depending on the pressure, the particular one illustrated being the 'duplex' packing, suitable for pressures of 150 lbs. and above. It consists of two parts, the lower A B termed 'block packing,' suitable for low pressures when used alone, combined with the upper cone packing C D. The block packing consists of two sections containing antifriction or whitemetal blocks which

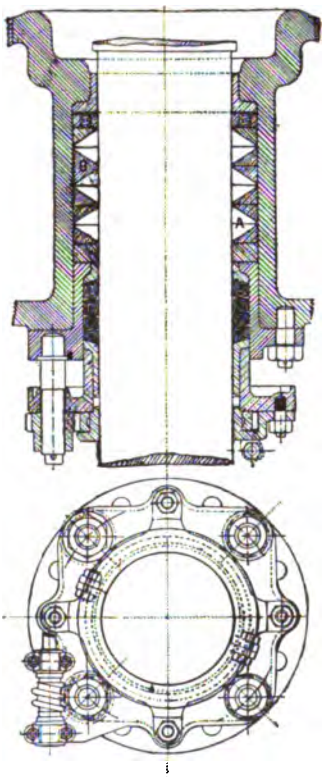


FIG. 217.

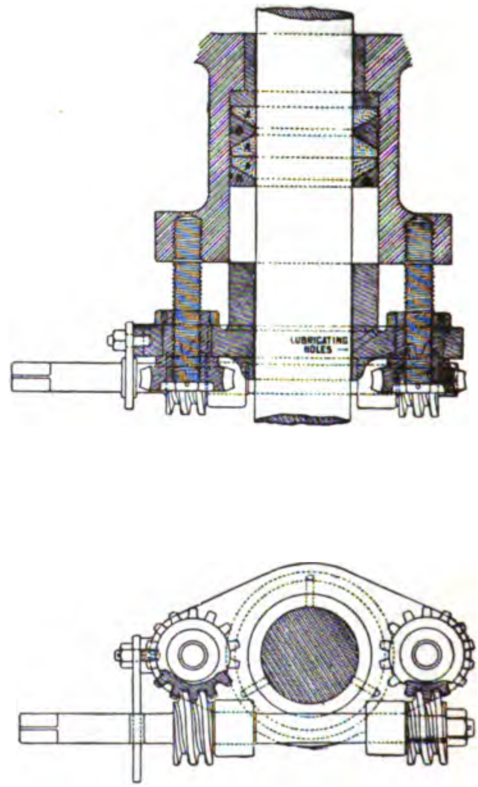


FIG. 218.

are held up to a bearing on the rod by springs. Each section, of which one is shown in the lower portion of the diagram, consists of two packing blocks, E, alternating with two guide blocks, F. The joints between the white metal blocks in one section are at right angles to those in the other section, thus 'breaking joint.' The springs simply keep the blocks in proper position when the steam is shut off, or when the piston rod is making its outward stroke, in which case the steam pressure is removed from the gland. When steam is applied, its pressure sets the packing, the design being such that by this means just sufficient pressure is applied to the packing blocks. There is a 'ball joint' at G, and at the other end, H, a follower ring and springs to take

up any longitudinal play, and by these means the steam tightness is maintained, even with slight displacements sideways of the rod. The upper part consists of white metal rings, J, contained in a bush which is capable of side motion, a conical joint and follower springs being fitted as in the 'block packing.' The part against which the follower bears is prolonged upwards beyond the bottom of the cylinder, so that any water of condensation can be led away by drain holes without passing between the rod and packing.

For very low pressures, as in low pressure cylinders where the

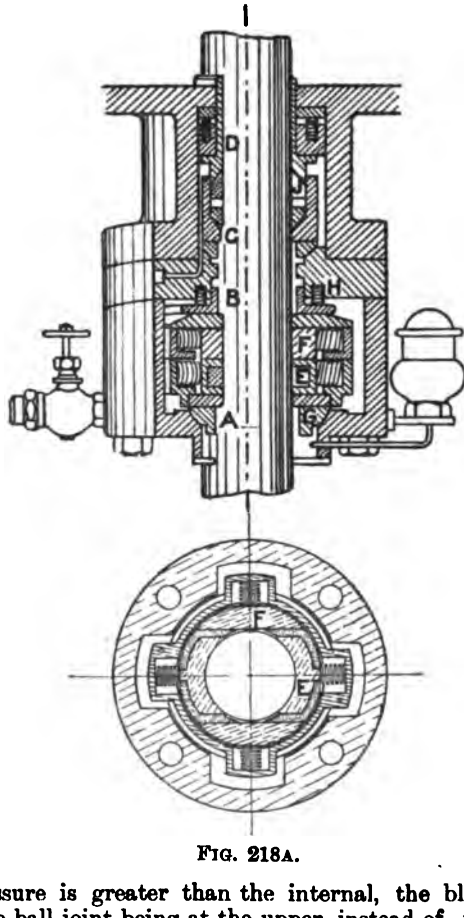


FIG. 218A.

external pressure is greater than the internal, the block packing is inverted; the ball joint being at the upper instead of the lower end, and the follower ring and springs at the lower end.

Screwing-up gear.—The nuts of the piston-rod, slide-rod, and other principal glands, when they are so constructed as to be kept tight by compressing soft packing, are fitted with toothed or worm gearing, so that they may be screwed up or slacked uniformly, and adjusted with safety when the engines are at work. Arrangements of this kind are shown in the preceding sketches, where it will be seen that for each of the

glands the screwing-up gear is fitted to the gland bearing on the soft packing, and is arranged to be screwed up with a right-hand motion.

Cylinder escape valves.—It is necessary that escape or safety valves should be fitted on the cylinders, so that in case of water accumulating from priming or condensation during the working of the engines, means of escape will be provided, to prevent excessive stresses being brought on the cylinders. These valves are generally ordinary conical valves, kept in their places by springs loaded a little above the maximum working steam pressure in the cylinder, and long enough to allow the valve to open a considerable amount without unduly increasing the load. Cylinder escape valves should be fitted with

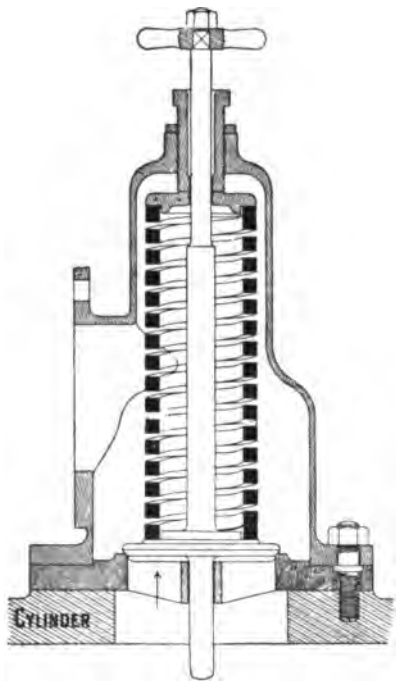


FIG. 219.

suitable guards and pipes to lead away the hot water and prevent the danger of people being scalded by its sudden escape. Fig. 219 shows such a valve. Provision is made for adjusting the compression of the spring and locking it in this position, and for turning the valve on its face. The guard and branch for conveying away water are shown in the sketch. The pipe attached is led to the bilge, but with its end in such a position that any water discharged can be readily seen.

The top of the high-pressure cylinder and the bottoms of all the main cylinders are usually fitted with escape valves, but they are often omitted from the tops of the intermediate and low-pressure cylinders, and occasionally from the high pressure. It is desirable, however, especially when the starting valves admit steam to the cylinders direct, that

escape valves should be fitted on the top of all the cylinders, and this is the usual practice.

Cylinder relief or drain cocks.—In addition to the escape valves, small relief or drain cocks, worked by levers from the starting position, are fitted at the lower ends of the cylinders, by means of which they can be cleared of water. They are generally fitted to discharge into the condensers when under way. In the Navy the high and intermediate-pressure drains are led to the auxiliary condenser, and the low-pressure to the main condenser. A branch pipe from a switch cock is also fitted to enable the water to be drained to the bilge when the engines are not working, and when starting. These drains were formerly led to the feed-tank, but owing to their noise, and for other reasons, this is not now generally done.

The levers for working the cylinder relief cocks should be led to a position close to the starting wheel or lever, and the handles should be in the same consecutive order as their corresponding cylinders to prevent mistakes. Where pipes from different cylinders or slide casings are led into a common pipe, non-return valves are generally fitted.

Slide jacket drains.—Cocks are also fitted for draining water from the lower parts of the various slide jackets, and these should be capable of being worked from the lower platform to insure rapid opening when required.

Indicator cock.—This is a three-way cock connected by pipes to either end of the cylinder, and having on the exit orifice a screwed socket to which the indicator may be attached. The hole in the plug of the cock is right-angled, so that the indicator may be put into connection with each end of the cylinder in turn, and the diagrams showing the work done on the opposite sides of the piston thus taken on a single card. A small hole in the shell of the additional cock fitted on the indicator enables the indicator piston to be put in communication with the atmosphere, to enable the atmospheric line to be drawn on the diagram.[1]

Auxiliary starting or pass-valves.—To facilitate the handling of the engines, in which steam of boiler pressure is admitted to one cylinder only, small auxiliary starting valves or pass-valves are fitted to the cylinders, worked by hand levers at the starting position. These valves take their steam from the main steam pipe, and they provide the means of admitting steam, by hand, to the cylinders. The valve fitted for this purpose may be one of two kinds, viz. auxiliary starting valves or pass-valves.

Auxiliary starting valves are those which admit steam direct to the top or bottom of the cylinders—i.e. between the slide-valve and piston. By moving the lever in one direction steam is admitted to the top of the piston, and by moving it in the reverse direction steam passes to the bottom of the piston. The levers are preferred to be so arranged that they move in the same direction as the steam admitted tends to move the piston, so that if, on looking at the crank, it is seen that it is required to descend to move in the correct direction, the auxiliary starting valve lever would be moved downwards. With the three-crank triple expansion engines, these starting valves would be fitted on the intermediate and low-pressure cylinders, starting valves on these

[1] See Chapter XXVI.

two cylinders being sufficient to enable the engines to be started when the high-pressure slide-valve is closed to the admission of steam.

Pass-valves are those by means of which steam is admitted to the receiver spaces, and not to the cylinders direct. They are fitted to the intermediate receiver, and also to the low-pressure receiver, either two entirely separate valves and levers being fitted, or one valve and lever, the motion of which in one direction admits steam to the I.P. receiver or steam space of I.P. slide-valve, while motion in the reverse direction admits steam to the L.P. receiver or steam space of slide-valve. By one of these means steam is admitted to the receiver required, and the steam proceeds to the top or bottom of the piston depending on the position of the slide-valve. Should the slide-valve of that engine be in such a position that neither port is open to steam, the pass-valve will not be able to operate on the piston, and steam to the other receiver must be admitted. Care must be taken in the design of the engines that the cut-off in the cylinders is sufficiently late that, when either crank is at or near the dead centre, one of the other slide-valves is always open to steam both for ahead or astern working.

Pass-valves require no mental consideration as to which direction the piston should be moved; but, on the other hand, they are rather slower in their action, as the steam has to fill the receiver as well as the cylinder space on one side of the piston, and at the same time that it increases the forward pressure on one piston it increases the back pressure on the preceding but smaller piston. On account of their simplicity in use and their freedom from error they are generally preferred. Fig. 221 shows such an arrangement, one valve and lever only being used, the motion of which in one direction admits steam to the I P receiver, while motion in the other admits steam to the L P receiver. A detail of A, the pass-valve itself, is shown in Fig. 220.

Receiver safety valves and pressure gauges.—The intermediate and low-pressure receivers convey steam to the respective cylinders, and as each of the latter is constructed and tested for a working pressure of steam much lower than that in its preceding cylinder, safety valves are fitted on each receiver to give warning should the pressure in the cylinder by any means exceed the proper amount. Such an excess of pressure would occur probably if the slide valve of the preceding engine were to get off from the cylinder face, and in starting the engines. These safety valves discharge their steam into the engine room, and so give warning to those in charge.

The pressure in these receivers is shown at the starting platform by pressure gauges, which are always fitted, and connected to the high-pressure, intermediate, and low-pressure *slide jackets*. With the ratio of cylinders used in the Royal Navy, for a steam pressure of 150 lbs., the safety valves on the intermediate and low-pressure receivers are loaded to 80 lbs. and 30 lbs. respectively, and for a steam pressure of 250 lbs. at the engines the loads are 130 lbs. and 45 lbs. respectively.

Steam cylinder jacket fittings.—To fill the jacket spaces around the cylinder barrels with steam, a steam pipe is led to a stop-valve on the main steam pipe. Before entering these spaces, in the case of the intermediate and low-pressure cylinders, the steam passes through *reducing valves*, by means of which the pressure, and therefore temperature, in the jacket can be regulated to that suitable for the

working pressures in the cylinders. Small *safety valves* are also fitted to the intermediate and low-pressure cylinder jackets to insure the desired pressure not being exceeded. These safety valves blow off into the engine room, and so attract attention when the proper pressure is being exceeded, either by failure of the reducing valve or other cause. Steam of boiler pressure can generally be admitted to the high-pressure jacket, but sometimes the steam supply is taken from the high-pressure slide casing so that the jacket pressure can never be above the

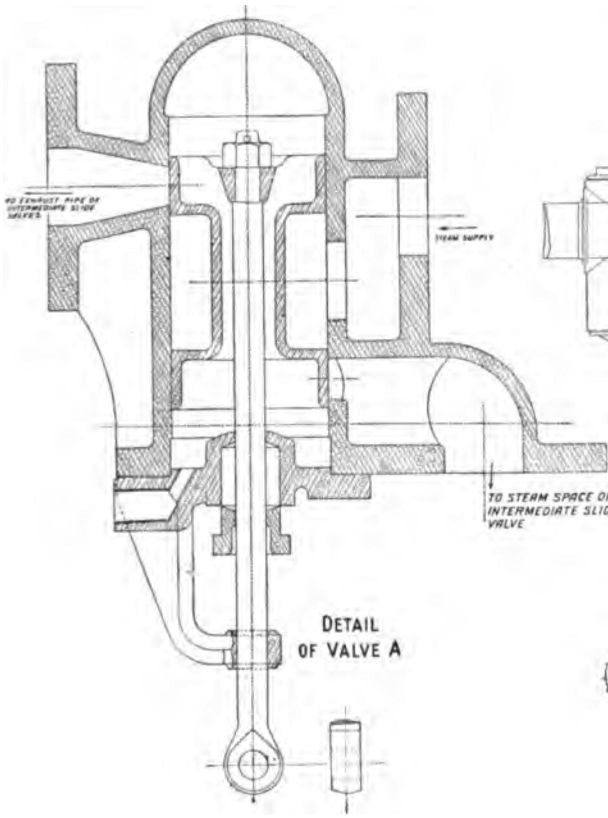


FIG. 220.

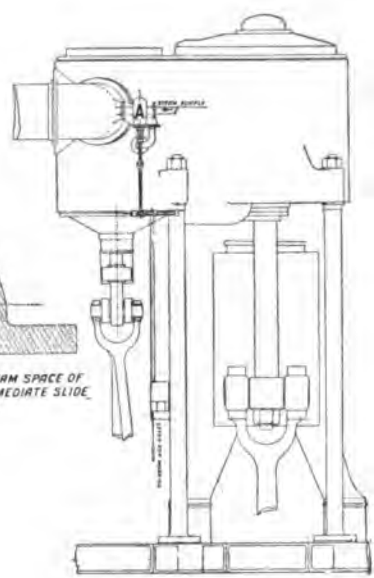


FIG. 221.

initial pressure in the cylinder. In the other jackets the reducing valves are adjusted so that the pressure is a little in excess of the maximum working pressure in the corresponding cylinders.

With steam of 150 lbs. boiler pressure, for example, the steam-jacket pressures in intermediate and low-pressure cylinders would be about 80 lbs. and 30 lbs. respectively. In the naval engines now building with 250 lbs. working pressure, it is 130 lbs. and 45 lbs. respectively. *Gauges* are fitted to indicate the pressure in each of the cylinder jackets.

To keep these steam jackets free of water, *drains* are led from their lower parts to the auxiliary condenser, or main condenser when an auxiliary condenser is not fitted. In the Navy in order to facilitate the adjustment of the drain-valves to the required amount so as just to remove the water of condensation, each of the drain-pipes is led to a water collector in sight of the starting platform. These collectors have glass water gauges and regulating drain-valves to enable the adjustment to be accurately made. The drain-valve should be so regulated that the collector remains always about half full of water, and when this is so we know that the cylinder jacket is properly drained and that no steam is being blown through the valve to waste.

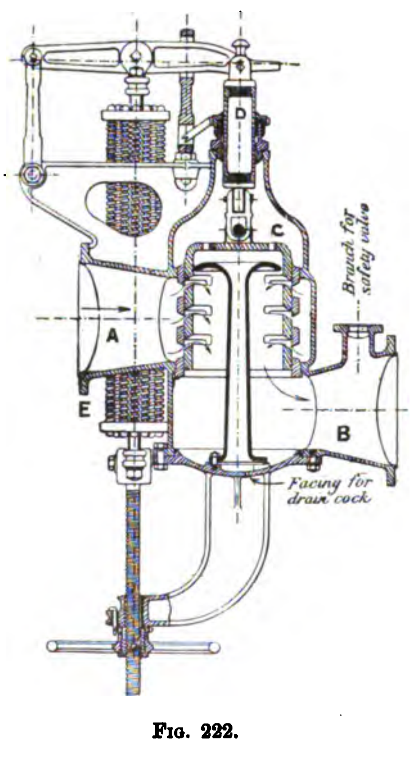


FIG. 222.

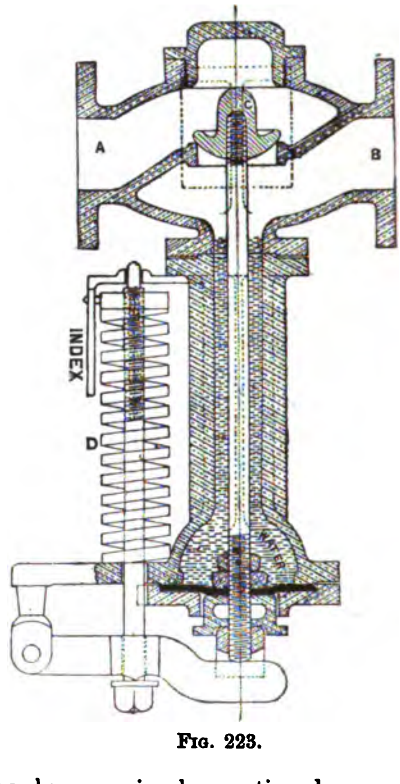


FIG. 223.

Reducing valves.—The reducing valves previously mentioned are fitted for several reasons. They insure that the pressure of steam supplied for some purposes, from a source at higher pressure, does not exceed the amount it may be safe or desirable to use, as, for example, in steam jackets. They are useful in maintaining steadiness of the steam pressure working an engine, although the steam pressure in the boilers may vary, which is important, for example, in the case of dynamo engines. They have also, for similar reasons, been largely used in the Royal Navy for the main engines of most of the vessels fitted with the higher pressures of steam and water-tube boilers.

Two common types of reducing valves are illustrated, in both of which the valve remains open till the pressure of steam on the reduced side attains a certain amount regulated by springs, when the valve closes. Fig. 222 shows the Belleville reducing valve fitted between engines and boilers of the earlier warships with Belleville boilers, in which there is generally a difference of 50 or 60 lbs. pressure between engines and boilers. In this case the tension of the springs E keeps the bell valve C open, but as the steam flows through from A to B, the pressure in B gradually rises, and as all parts of the valve are balanced, as regards pressure, except the area of the plunger D, the reduced steam pressure passing through the two holes at the top of the bell, acts on the area of the plunger, and when this force is sufficient to overcome the tension of the springs acting at the smaller leverage, the plunger D rises and closes the ports of the valve. The valve and seating are an easy fit so that this type of valve will not prevent the passage of small quantities of steam. It is generally used for the main engines where large volumes of steam have to be passed. A small safety valve is fitted on the reduced pressure side as a relief in case of any derangement of the gear. Pressure gauges are also fitted on each side of the reducing valve, placed in positions visible from the starting platform when fitted for the main engines. Experience in the British Navy indicates, however, that reducing valves are not absolutely necessary for the main engines when water-tube boilers are fitted, and these reducing valves are not fitted in new ships.

Fig. 223 shows another example, Auld's, in which the balancing of the valve and the closing force are entirely differently provided for. In this example the compression of the spring D, which can be regulated by a nut and screw, keeps the valve open. The higher-pressure steam enters at B, and this steam has no effect either in opening or closing the valve C, since the steam pressure on the bottom of the valve is exactly balanced by the pressure on an indiarubber diaphragm at the other end of the casing, over an orifice of exactly the same area as the valve. This indiarubber is shown in black in the figure, and is kept cold by an accumulation of water which condenses above it. The steam pressure in B having no effect, as soon as the pressure in A, acting on the top of the valve, reaches an amount sufficient to compress the spring still further, the valve closes and shuts off the supply of steam.

CHAPTER XX.

CONDENSERS, FEED-WATER FITTINGS, AND UNDER-WATER VALVES.

Composition of sea-water.—Sea-water contains about $\frac{1}{33}$ part of its weight of solid matter, the composition of which varies somewhat with the locality. Common salt or sodic chloride is always by far the principal constituent of the solid matter. Of this $\frac{1}{33}$ part of solid matter the following may be taken as the average composition :—

Sodic chloride or common salt	76	per cent.
Magnesic chloride	10	"
Magnesic sulphate	6	"
Calcic sulphate or gypsum	5	"

The remainder, 3 per cent., consists of small quantities of carbonate of lime and other substances, with a little organic matter.

Formation of scale.—Common salt, however, gives little trouble to the marine engineer, as, unless under extreme circumstances, it remains soluble in water at all temperatures. The density of sea-water may be increased very largely by concentration of the common salt, and its temperature be considerably increased, without any deposit of this material taking place. The principal scale-forming ingredient is the sulphate of lime or calcic sulphate, which is most troublesome when admitted into the boilers. Deposit is also formed by the sulphate of magnesia, although in a less objectionable form. The deposit from sea-water in boilers is very hard and difficult to detach, and consists of about 85 per cent. of sulphate of lime.

While common salt is just as soluble at high temperatures as at low, it is found that as the temperature of the water increases, a point is soon reached at which the sulphates become insoluble in water, and if admitted to a boiler in which the water is at, or above this temperature, they are precipitated in the solid form and remain in the boiler.

At a temperature of 280° to 295° Fahr., corresponding to a pressure of 35 to 45 lbs. of steam by gauge, the sulphate of lime becomes entirely insoluble, and this substance amounts to 5 per cent. of the solid matter contained in sea-water. As the temperature rises the other sulphates become insoluble, until at about 350° Fahr. or 120 lbs. absolute the sea-water is incapable of holding any sulphates in solution, and if any sea-water is admitted *a deposit necessarily takes place.*

These salts are also precipitated by increase of density from the evaporation of the water, even if the temperature remains about

212° Fahr. ; sulphate of calcium is thus deposited at a density of $\frac{3}{32}$. Common salt does not crystallise out till a density of about $\frac{8}{32}$ is reached. Page 118 explains the term 'density' in connection with sea-water.

Use of jet-condenser.—In the first marine engines the temperature of the steam was low, and the jet-condenser was used, in which the steam, after leaving the cylinders, enters the condensing chamber, into which a jet of cold sea-water is injected, which condenses the steam by *actual contact and mixing*. The feed-water, drawn from the mixed sea-water and condensed steam, was only a little fresher than sea-water (density about $\frac{26}{27}$ that of sea-water), hence a large quantity of sea-water was sent into the boilers, and to prevent gradual increase of density, and consequent deposits of salt and other substances, a portion of the denser boiler-water had to be blown away into the sea, and the loss made up by admitting a larger quantity of the salt feed-water. This was termed *brining* the boiler.

As, however, the *temperature was low* and the density was not allowed to exceed about $\frac{2}{32}$, the salts were held in solution fairly well, so that but little deposit was obtained even from the sulphate of lime, which is the first to be precipitated. The steam pressure of 35 to 45 lbs. or the density of $\frac{3}{32}$ could not be exceeded with salt water feed without deposit of salts, and when the pressures of steam, and consequently the temperatures, were increased, it was found impossible to prevent the deposit, and in fact the process of brining increased it, from the extra salt water necessary. With high pressures, also, the injurious effect of scale deposited on heating surfaces is increased, so that sea-water feed for high-pressure boilers is not practicable.

Introduction of surface-condenser.—The surface-condenser was therefore introduced, in which there is no mixture of the steam to be condensed with the cooling sea-water. The sea-water is pumped through or around a number of small tubes ; the other side of these tubes is in connection with the exhaust steam, so that the latter is condensed by the cold surface of the tubes, and the resulting fresh water is pumped away, to be returned to the boilers as feed-water. By its agency fresh feed-water is obtained for the boilers instead of salt water, and the use of high-pressure steam, retarded for many years by the use of the jet-condenser, has now become general. One advantage of the surface condenser is that the condition of the condensing water is of no importance as regards the feed-water, so that whether it is salt, muddy, acid, or otherwise impure, pure water is obtained for the boilers, provided the condenser is maintained in good condition, and no leakage is allowed to occur.

Details of jet-injection condenser.—This consisted essentially of a cast-iron condensing chamber, into which exhaust steam enters through eduction pipe, and a sea-injection valve, on its side, worked by levers from the starting platform, and in connection with the sea by means of a Kingston valve and cock. The water was admitted through a series of holes in the internal injection pipe to facilitate its mixing with the steam.

Air-pump of jet-condenser.—The sea-water and condensed steam must be pumped away constantly, and as sea-water contains air in solution, which may be liberated either by boiling it or reducing its pressure, when the sea-water enters the condenser the reduced pressure and increased temperature liberate the air, and unless pumped away regularly, as it cannot be condensed, it would accumulate and spoil the vacuum.

The pump used is called the *air-pump*, and was generally worked by a rod attached to the piston of the engine. By the action of the air-pump, with its foot-valves and head-valves, the mixed injection and condensed water, air, vapour, &c., are pumped into a space called the hot-well.

Feed-pumps, discharge pipes, &c.—From the hot-well the feed-pumps draw their supply for the boilers. These pumps were usually horizontal plunger pumps worked off the main pistons, and a large discharge pipe led to a self-acting main discharge valve on the ship's side to allow the surplus water not required by the feed-pumps to pass overboard.

Bilge-injection of jet-condenser.—To all jet-condensers an additional bilge injection valve was fitted, with inlet pipe leading to the bilge. With a serious leak in the ship and the bilges flooded, when the engines were at work, water could be taken from the bilge, instead of from the sea direct, and the air-pump utilised for pumping out the ship.

Sea and bilge injection for early surface-condensers.—In the surface-condenser, the cold condensing water is kept flowing past the cooling surfaces by means of a pump called the circulating pump, and in the early surface-condenser, before sufficient experience was

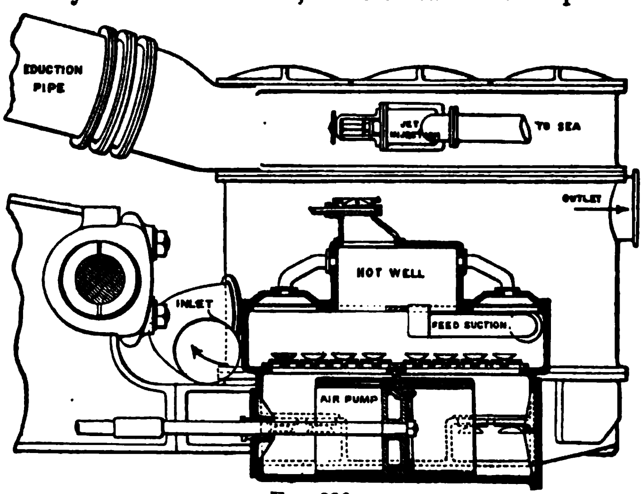


FIG. 226.

obtained of its action, arrangements were made for the admission, if necessary, of sea-water into the steam space of the condenser, thus converting it into a jet-condenser in case of failure of the circulating pump. Fig. 226 shows an elevation of an old surface-condenser showing position of jet-injection valve and section of air-pump. This jet-injection valve was, however, soon abandoned, and is not now fitted.

Arrangements were also made for admitting water from the bilge into the steam space of the condenser, so that the air-pump would assist in pumping water out of the ship. The circulating pumps of such condensers were generally of the reciprocating kind worked off the main pistons, but on the adoption of separate centrifugal circulating pumps for this purpose, described later, which also enabled considerable quantities of water to be pumped from the bilge if

required, these bilge injection arrangements were abandoned. The surface-condenser now fitted contains neither sea nor bilge injection.

The surface-condenser.—This consists of the 'condenser barrel or casing,' generally of cylindrical shape, having a flat plate at each end called a 'tube plate.' Between the tube plates are fitted a large number of small thin brass tubes, which are kept cold on one side by sea-water, supplied constantly by the 'circulating pump.' The exhaust steam is conveyed to the upper part of the condenser, and is condensed on the other side of the tubes, the condensed water and any air being withdrawn by the air-pump.

The tubes may be placed either vertically or horizontally. With vertical tubes the steam is generally passed through them, and the water circulated around them, while with horizontal tubes the steam is generally outside the tubes. The exact plan adopted, however, depends principally on convenience of arrangement. If the steam passes inside the tubes, the water being in the space outside, the grease left on the tubes by the steam can be readily cleaned off by passing brushes through them, and there is a great advantage in being able to quickly localise the position of leaky ferrules or tubes, but with steam outside the only method of effectually cleaning them is by filling the condenser with a solution of potash or other material and boiling it. In the Navy, with large powers two condensers are fitted on each side of the ship, which enables one half to be shut off if any defect occur, so that the latter can be remedied while the ship is still able to steam at a fairly high power with the other condenser.

Condensers with steam inside the tubes have the disadvantage of holding a larger quantity of water than those in which the water passes through the tubes, and this weight of water has to be added to the total weight of machinery when the engines are at work. The number of joints which are liable to leak and affect the vacuum is also larger with this system, while the condenser casings are in contact with the circulating water, which causes them, when made of cast-iron, to corrode and decay. The system of causing the water to pass through the tubes, the steam being in the space surrounding them, is the more general. The casings are somewhat hotter in consequence of containing steam instead of water, but the heat can be prevented from appreciably affecting the engine room by lagging.

Figs. 227 to 230 show two plans of surface-condenser, viz. the horizontal and vertical varieties. Referring to Fig. 228, in which there are two exhaust pipes, the steam enters at the orifices marked A, and is withdrawn, when condensed, through the orifice B by the air-pump. The circulating water enters at C, and is confined by the diaphragm D to the lower half of the tubes, and, having traversed these tubes, it returns through the upper half of the tubes, being finally discharged to the sea through the pipe E. In the vertical condenser, shown in Fig. 230, the same letters apply, the circulating water being forced to traverse the whole of the condensing surface by means of the diaphragm D indicated in sketch. In this type it is important that these directing plates should be fitted, to ensure the proper efficiency of the cooling surface.

Details of the condenser.—The condenser casing is in the Navy usually of brass, and is either cast, as in Fig. 230, or built up of brass plates riveted together and stiffened by angles, as in Fig. 228. As this material

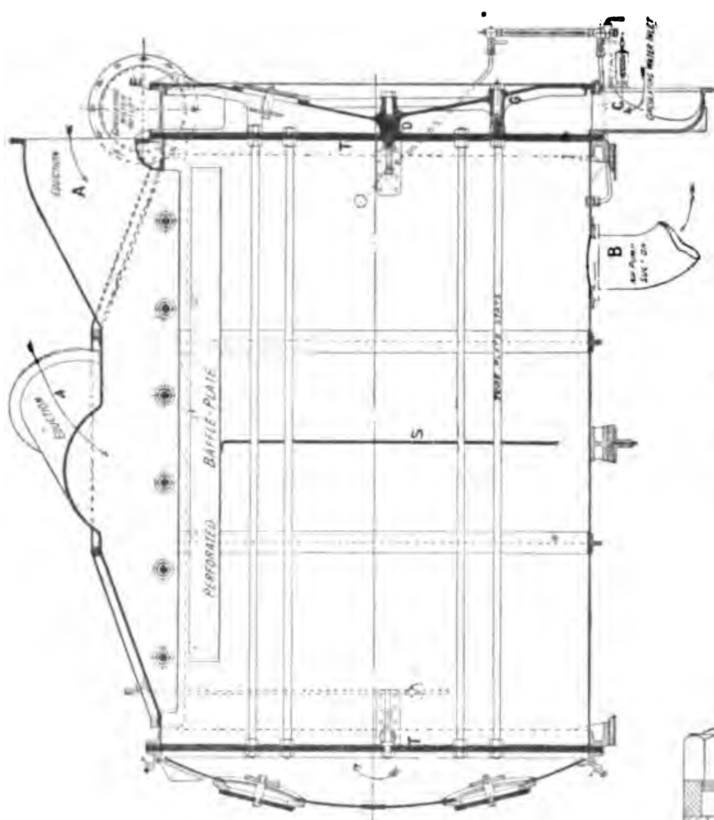
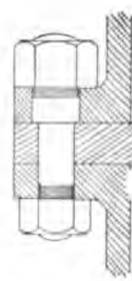


Fig. 228.



ENLARGED VIEW OF TUBE PLATE JOINT

Fig. 229.

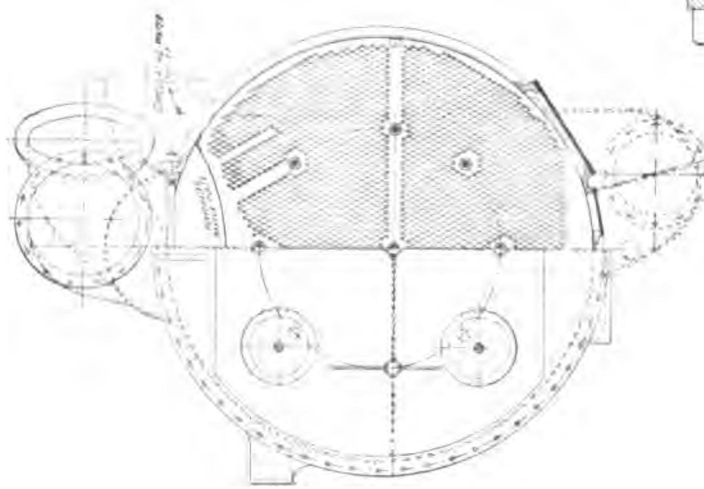


Fig. 227.

is free from any waste due to corrosion, the casing may thus be made thin and light. At the ends of the condenser casing are the tube plates T T. Between these tube plates a large number of small tubes are fitted. For clearness these are omitted in the sketches, but a few are indicated at the top of the section part of Fig. 227. These tubes are generally

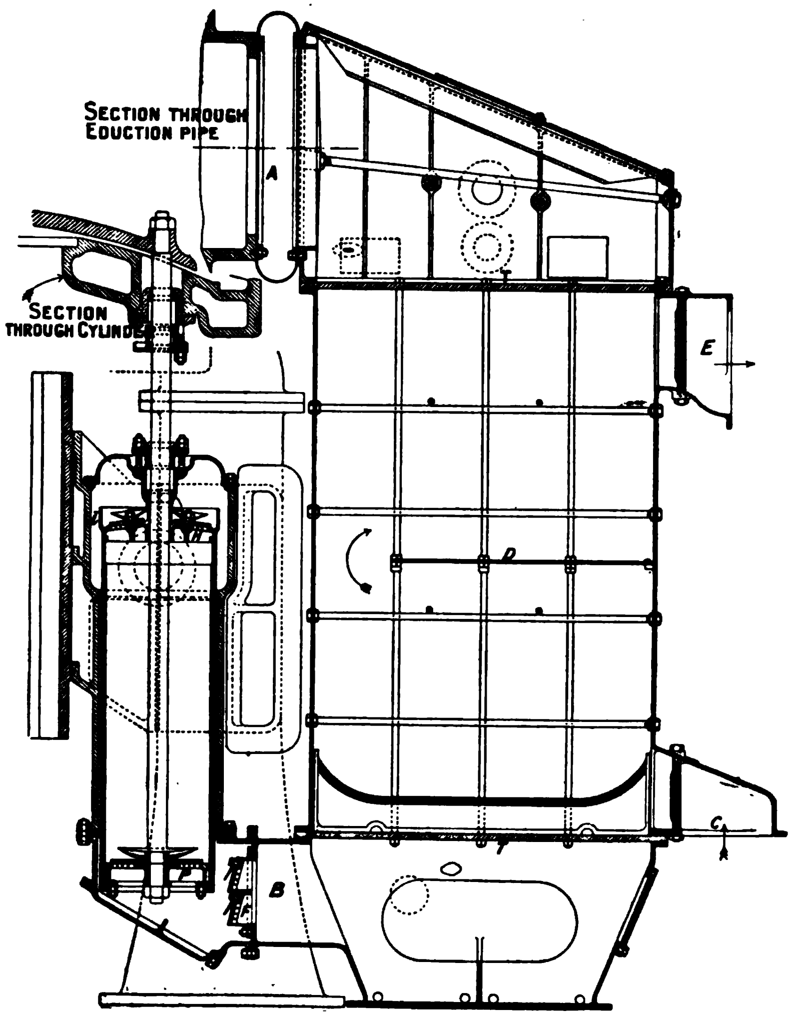


FIG. 230.

$\frac{5}{8}$-inch diameter, and pitched not less than $\frac{3}{3}\frac{1}{2}$ inch apart, to allow sufficient material for the gland. They are zigzagged so as to occupy as small a volume as possible, and are made of brass, about $\frac{1}{20}$ inch thick, of a composition consisting of not less than 70 per cent. of copper and not less

than 1 per cent. of tin, the remainder being zinc. The small quantity of tin is added to prevent galvanic action. The brass for the condenser casing and the tube plates is also made of as near the same composition as possible, or the tube plates are often made of Muntz metal.

The tube plates for large condensers are now made 1 inch thick, in order to provide sufficient depth for insuring a proper watertight attachment for the tubes; for not only must the tubes be watertight, but they must also be free to expand and contract with the changes of temperature they necessarily undergo. The plan of fitting generally used, and invariably used in the Navy, is to form small screwed stuffing-boxes in the tube plates, and provide screwed ferrules, fitting over the tubes, to tighten tape packings around the tubes at the bottoms of the stuffing-boxes. The ferrules are generally made with small internal projections or flanges at their outer ends, to prevent the possibility of the tubes slipping out of place. This is desirable in all condensers, at each end, but is essential for the lower ends of vertical tubes. These projections are invariably fitted in naval condensers. A sketch of the tube attachment is shown in Fig. 231. The tightness of the tube ends is very important, for should leakage occur the sea-water obtains access to the feed-water, is pumped into the boilers, and forms deposits.

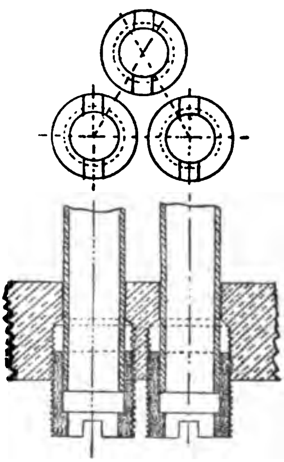


FIG. 231.

The tubes of condensers vary considerably in length up to about 14 to 15 feet, the general length in large condensers being from 8 to 10 feet. For new British war ships the condenser tube length is standardised, and they are either 5 feet or 10 feet long over all. With long tubes, supporting plates, S, Fig. 228, are fitted with holes in them to permit the passage of the tubes. These prevent bending of the tubes, and disturbance of the stuffing-box joint at the end of the tube. The tube plates are strengthened by a few stay-bars as indicated.

The condenser covers, G, are attached to the tube plates by bolts fitted so as to be independent of those securing the tube plate to the condenser casing, so that the cover joint may be broken without disturbing the joint between tube plate and condenser barrel. This is often effected by fitting collar bolts, as shown in Fig. 229, in which case it will be seen that any bolt can be renewed without taking off the cover. Doors are fitted in the covers for examination of the glands, and on the condenser barrel for examination of the outsides of the tubes.

The area of the condensing surface in surface-condensers was formerly made about the same as that of the heating surface of the tubes in the boilers, but experience shows that considerably less than this is sufficient, and in most modern ships with triple expansion engines the area of cooling surface is only from 1 to 1·25 square feet per I.H.P.; 1 square foot per I.H.P. at full power has been found in later war-ships to be sufficient. With steam turbines, however, where a higher vacuum can be advantageously utilised than with the reciprocating engines, an

increased cooling area is given, about 1·2 sq. ft. per I.H.P. in the latest turbine war-ships. In torpedo-boat destroyers, where weight is of extreme importance, the area is sometimes as low as ¾ sq. ft. per I.H.P.

Air-pump of surface-condenser.—In surface-condensers the only water that accumulates in the condensing chamber is that from the condensation of the exhaust steam, but a small amount of air is always present, either liberated from the feed-water or due to leakage. An air-pump is therefore also necessary with surface-condensers, although it can be made very much smaller than that of the old jet-condenser.

The air-pump is of the reciprocating kind, and with vertical engines it is almost invariably fitted as a vertical single-acting bucket pump. With many horizontal engines, where a vertical motion can be conveniently obtained for the air-pump, this vertical variety of pump is also often fitted. With horizontal engines generally, however, the air-pump is also horizontal, and worked directly off one of the pistons or crossheads. In this case it is generally a double-acting solid piston pump similar to that illustrated in Fig. 232, but the vertical bucket pump is much more efficient.

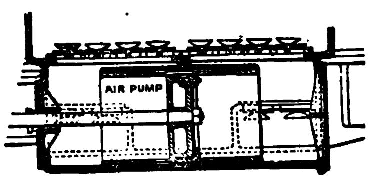


FIG. 232.

Vertical air-pump.—The vertical air-pump is illustrated in Figs. 230 and 233. It consists of a reciprocating bucket or piston, P, fitted with orifices covered by non-return valves, which move in a cylindrical barrel, at each end of which are fitted covers or seatings, F and H, also provided with orifices and non-return valves. These three sets of valves lift vertically, and only allow passage of water or air in the upward direction. The suction pipe B communicates with the bottom of the steam space of the condenser. The valves at the lower end are called 'foot' or 'suction valves,' those in the moving bucket are called the 'bucket valves,' while those at the top are the 'head' or 'discharge valves.' A large door is fitted in the air-pump barrel, so that the bucket and foot-valves can be examined without removal of the head-valves and cover. An air vessel is usually fitted above the head-valves and an escape-valve on the cover, this latter being for use in case an abnormal load is brought on the pump, such as when the engines are inadvertently started too quickly when there is a considerable amount of water in the condenser.

Action of the pump.—During the up stroke of the bucket a partial vacuum is formed between the bucket and the foot-valves, which causes the foot-valves to be opened by the slight excess pressure in the condenser, and water and air to enter the barrel. On this stroke, also, water and air above the bucket is forced out through the head-valves, H, into the discharge pipe, the non-return valves in the bucket being closed by the pressure above them, and preventing return of the water. On the down stroke commencing, the foot- and head-valves close, and the bucket descends through the air and water below it, the bucket valves being forced open and allowing the air and water to pass through to the upper side. On the next up stroke this air and water is discharged, and a fresh supply enters the barrel from the condenser.

Amount of vacuum possible.—It should be noted that there must be an excess pressure in the condenser beyond that in the air-pump barrel, sufficient to lift the foot-valves, so that even with the most efficient air-pump of this construction, a perfect vacuum in the condenser is not

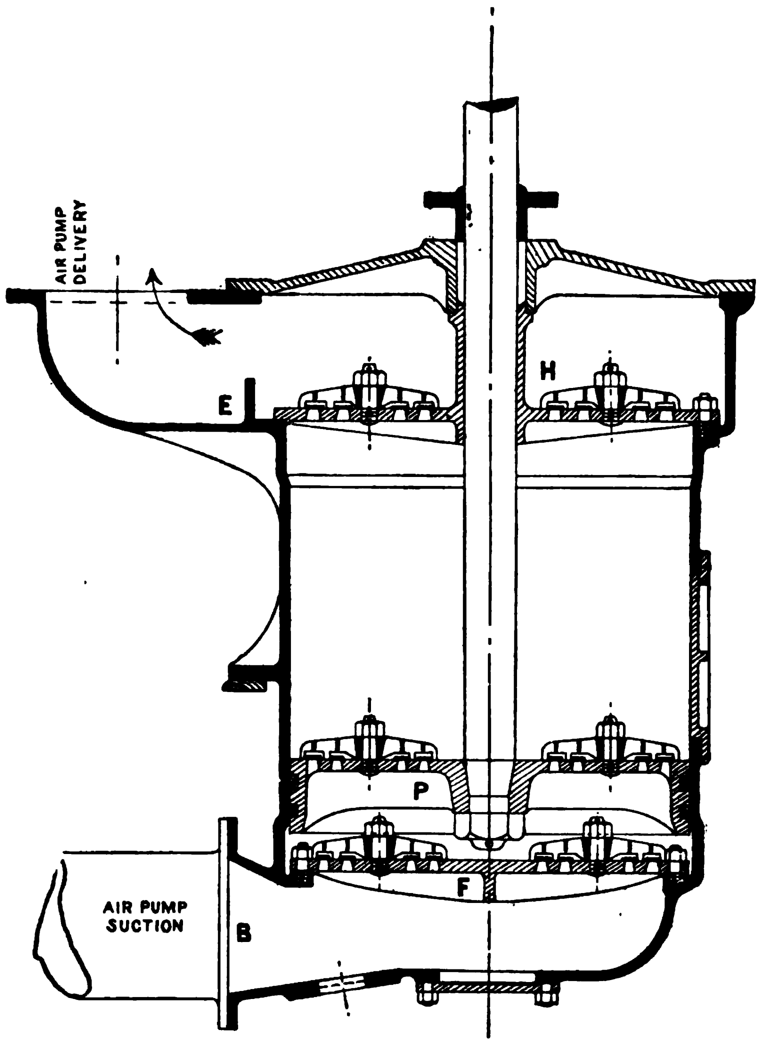


FIG. 233.

possible. The amount of vacuum possible in the condenser depends on the weight of these foot-valves, and also on the perfection of the vacuum the air-pump creates in its pump barrel, which latter, assuming the air-pump bucket to work airtight in the barrel, depends very greatly on the

amount of clearance spaces there are between the bucket and the foot-valves; the less this clearance is, the greater the vacuum possible, so that the clearance spaces should always be made as small as possible. With the usual construction the clearance spaces in vertical air-pumps are less than in the horizontal ones, as in the former the bucket can be more easily arranged to travel very close to the foot-valves at the bottom of the stroke. Another feature in which the vertical pump is superior to the horizontal exists in the liability of the air-pump rod of the latter to leak and impair the vacuum in the pump, whereas in the vertical pump this is of little importance. Any pockets and spaces in the pump chamber where air may collect should be carefully avoided, for if any accumulation of air exists, a good vacuum cannot be obtained.

When all the parts of the condenser and engine are in good order, the amount of air present is very trifling. On the up stroke of the pump, therefore, we have generally present, above the bucket, water and water vapour. On being compressed, this water vapour returns to the liquid form, so that there is often some shock when the solid water strikes the head-valves and commences to be discharged. To avoid this a small adjustable non-return pet valve is generally fitted to the barrel just below the head-valves, which enables a small quantity of air to be sucked in on the down stroke, which acts as a cushion above the bucket on the up stroke, and so reduces shock. As the bucket also strikes the water suddenly on the down stroke, this pet valve is sometimes fitted a little below the bucket at the top of its stroke, so that a small amount of air is admitted below the bucket to act in a similar manner.

The barrel and bucket of the air-pump and the seatings for the foot- and head-valves are made of gunmetal, and the air-pump rod is made either of the same material, or preferably of some rolled bronze or brass, such as rolled naval brass or manganese bronze.

In the horizontal double-acting air-pump (Fig. 232), where the water only passes through two sets of valves, it is very important that the head-valves should be kept always covered with water, as this improves their action; the discharge from the air-pump is arranged to insure this. In the vertical bucket-pump this is also desirable, and the sketches show the ledges, *l*, Fig. 230, and *x*, Fig. 233, fitted for this purpose.

Air-pump valves.—Air-pump valves are either made of vulcanised indiarubber, or of sheet metal. Indiarubber, if used, has to be specially prepared to resist the action of the mineral oil used for the lubrication of the cylinders and slide-valves, which soon destroys ordinary indiarubber. When used for naval vessels, the indiarubber for air-pump valves contains an amount not exceeding 70 per cent. of oxide of zinc to enable it to resist the action of the oil, while sulphur is present to an amount not exceeding $1\frac{3}{4}$ per cent., the remainder being best caoutchouc, with no other ingredients. Air-pump valves are, however, now generally made of thin sheet metal, of which there are numerous varieties. Many of these give excellent results in practice. They can be made very light, are not affected by grease if occasionally cleaned, and last a very long time.

The orifices covered by the air-pump valves, whether of indiarubber or metal, require to be divided into small spaces by gratings, as in the plan Fig. 237, so that the unsupported area of valve is not too great, while

a brass guard secured to the seating by a stud and nut must be fitted to regulate the lift. Figs. 234 to 236 show various patterns of metal valves, and Figs. 237 and 238 examples of indiarubber valves. In some examples the seating is a separate casting bolted in position, as in Fig. 238.

Vertical air-pumps are generally worked by rocking levers and links, as shown in Fig. 192, from one of the piston-rod crossheads, generally the low-pressure, arranged so as to reduce the speed by about one-half and keep the speed of the bucket moderate.

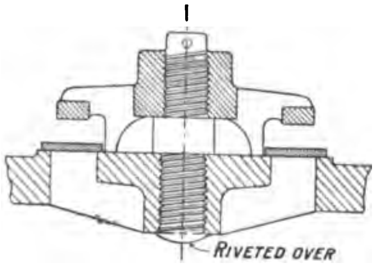


FIG. 234.

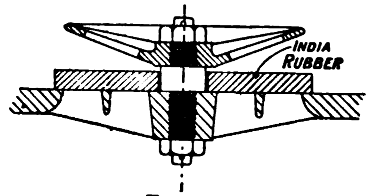


FIG. 237.

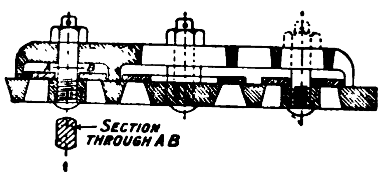
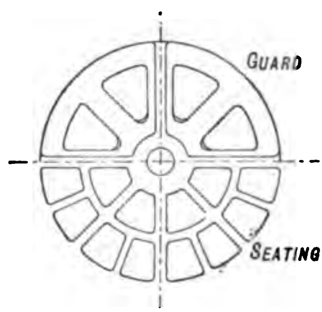


FIG. 235.



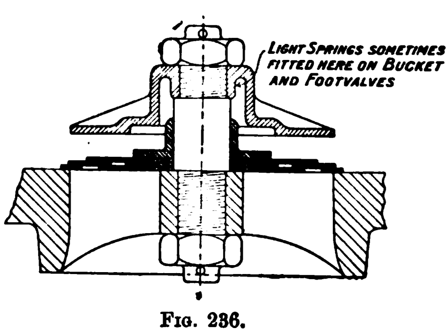


FIG. 236.

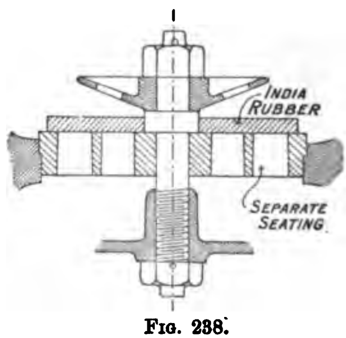


FIG. 238.

In the engines by Messrs. Humphrys, Tennant & Co., and occasionally by other firms, the vertical air-pumps are worked as in horizontal engines, from the piston of the engine direct, as shown in Fig. 230, the speed of bucket being the same as that of the engine piston. The attachment to the L.P. piston is indicated in this sketch.

Independent air-pumps.—In the most modern engines of high power both in the Navy and mercantile marine, especially in those

designed to work at high speeds of revolution, the air-pumps are not attached to the main engines but are worked by separate auxiliary engines, the main engines being employed for propelling purposes only. This enables the speed to be regulated independently of the main engines, and a vacuum to be maintained in the condensers whether the main engines are working or not, which greatly facilitates starting large engines. It also relieves the main engines of a considerable amount of complication, and the auxiliary condenser and the auxiliary air pump, together with their piping, can usually be dispensed with. Such pumps, however, to work satisfactorily must be well designed.

It should be noted that as the work done by the air pump piston increases at the ends of the stroke when the water is being forced through either the head valves or the bucket valves, the usual method of working the slide valve by an eccentric cannot be adopted, as this motion gives a reduced steam pressure at the ends of the stroke. The motion and setting of the slide valves have to be such that a full head of steam pressure is given at the ends of the strokes.

Weir's independent air-pump.—Messrs. Weir, of Glasgow, supply for marine purposes a type of independent air-pump which is very reliable and efficient in its action, giving a high vacuum with a steady motion, and is easily regulated to any desired speed.

In these pumps the water and air end is of the usual single-acting type, with foot, bucket and head valves. The barrels are entirely of gunmetal with gunmetal buckets and valve seats, manganese-bronze pump rods and Kinghorn valves with gun-metal guards. The steam cylinders rest on a cast-iron entablature supported by steel columns, and the steam valve chest is placed between the cylinders, with an arrangement to give steam to both. The pump rods are connected by links to rocking beams, from which the auxiliary valve in the steam chest is actuated, and this auxiliary valve operates the two main slide valves.

Blake independent air-pump.—This type of air-pump is similar to the preceding and is fitted in many vessels, including several United States warships; they work at slow speed and give excellent results.

A drawing showing this pump is given in Fig. 238A. It consists of two cylinders and two single-acting vertical air-pumps of usual construction, the rods of the two pumps being connected by a rocking beam so that when one pump is at the top of its stroke the other is at the bottom. The slide valves of the steam cylinders actuating the pumps, are worked by a separate small auxiliary steam cylinder and piston A, placed horizontally in front of the two main cylinders. The piston rod of this small cylinder works the two slide-valves of the main cylinders, one on each side, by means of a system of levers working inside the main slide casing and shown below the horizontal cylinder. The slide valve of the auxiliary cylinder is actuated from the shaft of the rocking beam by the small crank B and bell crank C, adjustable collars being fitted on the auxiliary valve-rod, which enable the travel of the small valve to be regulated so that the pump works with a full stroke of the plunger and at any speed desired. Small cushion valves, D, are also provided on the main cylinders for adjustment purposes. On the trials of some U.S. warships these pumps worked when at full power at fifteen double strokes per minute, maintaining a steady

vacuum of 25 inches, the power indicated in their cylinders being only $\frac{1}{600}$ of that of the propelling engines, and the pump plungers sweeping through $\frac{1}{72}$ of the volume swept through by the L.P. pistons.

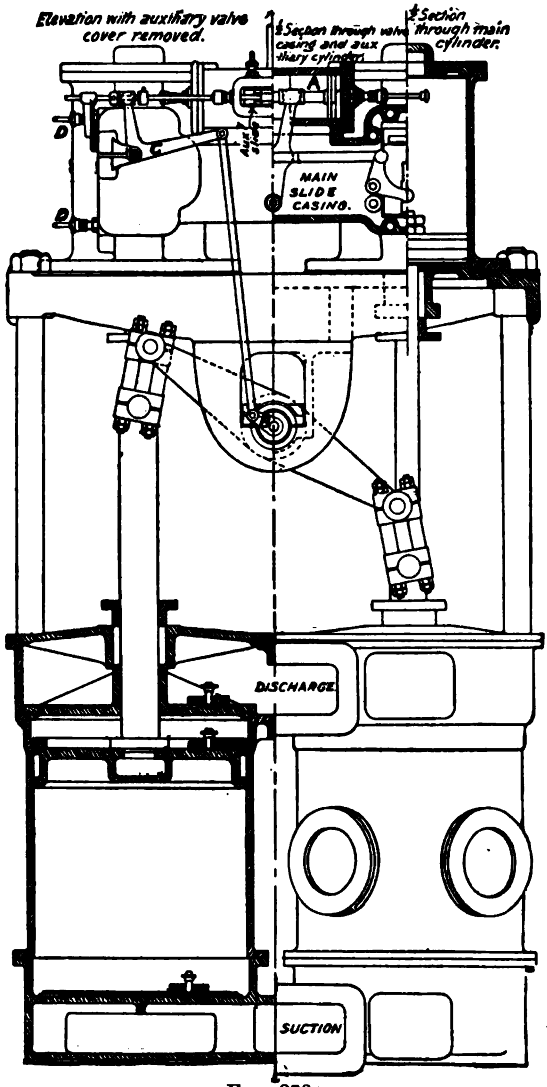
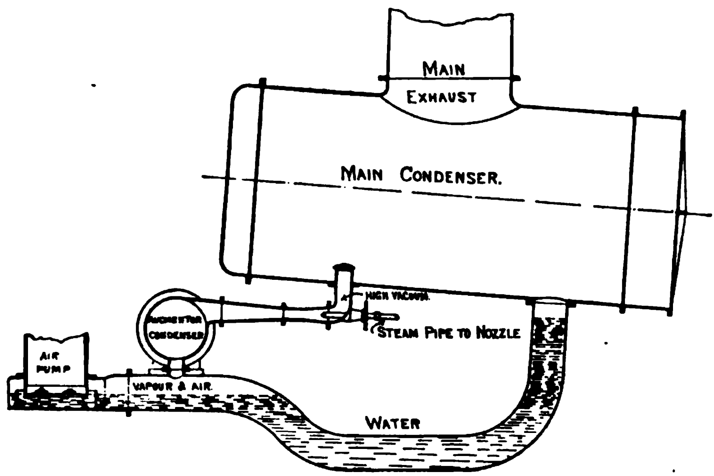


FIG. 238A.

Dry air pumps.—In cases where a very high vacuum can be utilised with corresponding economy, the main air pumps can be kept of moderate dimensions by supplementing them by what is termed a 'dry

air pump' i.e. a pump which is connected to the air pump suction pipe at such a height that it pumps vapour only, leaving the condensed water to be pumped by the main or wet air pumps. The latter pumps must of necessity be worked slowly, but the dry air pump can be worked at a fairly high speed of revolution and hence need not be large. By the combination of the two a high vacuum can be obtained, and this is very necessary with steam turbine installations where economy depends largely on its attainment.

Weir's dry air-pump is made of gunmetal, and arranged vertically over the steam cylinder, the piston rod of which is prolonged, and carries the air pump piston. The pump itself is single-acting, and is fitted with light bronze valves and springs. A cooling water connection is made on the upper part of the chamber, above the air pump piston, and is drained from the bottom of the pump to the condenser. An



VACUUM AUGMENTER.

FIG. 238B.

annular suction passage with openings surrounds the pump barrel, and this takes its suction from the upper portion of the air-vessel in connection with the wet air pump, and the discharge leads from the upper part of the barrel. The engine is of the single-acting enclosed type, with fly-wheel and governor, and has large bearing surfaces. The cranks are of forged steel, with crank pins and bearings hardened and ground, and in this case the valve can be worked by an ordinary eccentric.

Parsons' augmenter condenser.—A similar device for adding to the vacuum ordinarily obtained in the usual air pump is sometimes fitted in lieu of the above by Mr. Parsons. It is shown in Fig. 238B, and consists of a small supplementary condenser below the main condenser, with a small steam jet and contracted nozzle discharging air and vapour into the supplementary condenser, which is found to increase the vacuum. The air and vapour are compressed to about half

their bulk by the steam jet before entering the augmenter condenser, in which the air is cooled and vapour partly condensed and the whole then proceeds to the main air pump. The main suction pipe in the air pump is bent as shown in order to form a water seal. The augmenter condenser surface amounts to 2 or 3 per cent. of that in the main condenser and the consumption of steam in the jet amounts to about 1 to $1\frac{1}{2}$ per cent. of the total used. The whole arrangement is found to give an increased vacuum.

Feed-tanks.—The air-pump generally discharges its water through a pipe into a tank in the engine room called the 'feed-tank,' from which the feed-pumps draw their water for supplying the boilers. This tank should be of ample capacity, so that the feed-water may have space in which to accumulate when not immediately required for the boilers. Its capacity is generally at least equal to from four to five minutes' supply of feed-water at full power, and it provides a space in which the feed-water is at rest for a time, so that any air contained is more readily liberated, and passes off through a pipe which is open to the atmosphere. An overflow pipe, discharging into the reserve fresh-water tanks, is fitted to the feed-tank, with an internal pipe led to the lower part of the tank, so that if water is discharged from the feed-tank it comes from the bottom of the tank, and any grease floating on the surface of the water does not pass into the reserve fresh-water tanks. A small pipe is fitted to prevent syphoning. An additional overflow pipe with valve is fitted, so that any greasy surface water may be discharged into the bilge, except when a grease filter is fitted, in which case this overflow pipe is often omitted. A glass water gauge and zinc slabs are also fitted, and the feed-tanks of each engine room are connected by a pipe running between the two engine rooms, a shut-off valve being fitted in this pipe, worked from either engine room.

Hot-well tank and pump.—Some naval vessels are fitted with a tank, pump, and grease filter between the air-pump and the feed-tank. It is important that the feed-water should be freed as much as possible from grease prior to entry in the boilers, and it is desirable that any feed-water filters should be fitted to filter the water before admission to the feed-pumps, rather than on the discharge pipes of the feed-pumps. A grease filter between the air-pump and the feed-tank might, when dirty, bring too great a strain on the air-pump, so that, in several recent ships, the air-pump is allowed to discharge into an intermediate tank called the 'hot-well tank,' from which it is discharged by a pump fitted for this purpose called the 'hot-well pump.' This pump takes the feed-water from the hot-well tank, and discharges it through a feed-water filter into the feed-tank, from which tank the feed pumps draw water in the usual manner.

In the event of the feed-water filter becoming clogged up with grease so as to unduly increase the pressure required to be exerted by the hot-well pump to force it through the filter, an escape valve and pipe are generally fitted, so that under these circumstances the water lifts the escape valve and is discharged direct to the feed-tank. An overflow pipe is fitted to the hot-well tank or air-pump discharge pipe, so that in the event of the hot-well pump failing to act, the water will accumulate in the hot-well tank and overflow into the feed-tanks. The hot-well pump is of the ordinary reciprocating variety, generally

fitted with a float in the hot-well working the steam valve of the pump, so that a constant level is maintained in the hot-well. Details of the feed filters are described in Chapter XXVIII.

Reserve fresh-water tank and feed make-up arrangements.—The practice for many years was to fit a supplementary feed-pipe, consisting of a small pipe connecting the steam and water sides of the condenser, and fitted with a valve, so that any losses of feed-water resulting from leaks, safety valves blowing, &c., could be made up by admitting a quantity of the circulating water to the steam space by this supplementary feed-pipe. With the higher pressures of steam, however, it is found necessary, and is also economical, to prohibit this use of salt-water for auxiliary feed purposes. Arrangements are therefore supplied to enable fresh water to be used under these circumstances.

Reserve fresh-water tanks are supplied, the double bottoms of the vessel being generally utilised for this purpose, and these tanks form a reservoir into which fresh water can be placed from the shore, or into which the distilling apparatus on board can discharge, so that when extra feed-water is required to make up losses, the water in these reserve tanks can be drawn on. Instead of the old pipe and cock between the steam and salt-water sides of the condenser therefore, modern vessels are fitted with a pipe from this reserve tank to the steam space of the condenser, or the air-pump, with a valve conveniently situated for regulating. Losses of feed-water are made up by opening up this connection and admitting the extra fresh water to the condenser whilst the main engines are working. A connection is also made to the auxiliary feed-pump suction, which enables this pump to draw water direct from the reserve tanks.

Zinc slabs are fitted to prevent corrosion, also means for ascertaining the height of water, and air pipes to allow air to escape from the compartment, or to enter, when the tanks are being filled with water or being pumped out. A valve is also fitted so that these tanks can be connected to one of the Downton pumps, and emptied by this means if required.

Advantages of separate circulating pumps.—The cooling water of a surface condenser is generally circulated, i.e. pumped from the sea and returned again to the sea, through the condenser, by means of a *centrifugal pump* worked by an independent auxiliary engine. This plan is universal in the Navy, and is superior to a reciprocating pump worked direct from the engine piston, for with the separate engine the circulating pump can be kept working and the condensers kept cool when the main engines are stopped, whilst in the case of the pump worked off the main engines circulation ceases. In many cases with reciprocating pumps worked from the main engines, it has been necessary to fit suction pipes from the condenser casings to one of the auxiliary pumping engines, to prevent the condensers getting hot before starting the engines. It is also difficult when the pumps are worked by the main engines to vary the amount of circulating water if required, the speed being necessarily the same as that of the main engines. Often at the highest and lowest speeds the pumps do not work so efficiently as at moderate speeds, and the vacuum is consequently decreased. These pumps are also less available for pumping

out the ship in case of emergency, as they cannot be worked except when the ship is under way.

The centrifugal circulating pump.—Centrifugal pumps are very useful when large quantities of water have to be pumped with a comparatively small lift, as is the case in marine engine condensers, in which both the inlet and outlet orifices are generally below the surface of the water, and consequently the only work the pump has to do is to overcome the friction of the passages and keep the water in motion. They work very smoothly, have no valves, and have the further advantage that if they should be started before the outlet valve on the ship's side is opened, there is no fear of injury to the condenser, as the pump will only churn the water and not bring great pressures on the passages. When the discharge orifice in centrifugal pumps is not at the highest part of the casing, it is necessary to fit an air-cock at the top to let off the air and prevent accumulation, as the pump must be kept full of water to insure its efficient action, the presence of air interfering with its working.

Figs. 239 and 240 show details of the centrifugal pump and engine for circulating purposes. The pump consists of an impeller, wheel, or fan, revolving inside a casing, B. The impeller generally consists of a central web, A, guiding the incoming water, with two side plates, C C, gradually approaching each other as they near the circumference, and between which run a series of curved vanes, D, from the boss to the circumference. These vanes are curved away from the direction of rotation as they proceed from boss to circumference. The water enters the central part of the impeller from the inlet pipe E, and is thrown by the rapidly revolving vanes D, outwards and around into the casing B, which surrounds the circumference of the wheel. The direction of rotation is indicated by arrows in the sketch. The casing B is of gradually increasing area, and leads to the delivery pipe F, along which the water is forced by the centrifugal action. It thence proceeds to the condenser, where, after traversing the tubes, it is again discharged overboard.

The impeller and casing are made of gunmetal, and the spindle is either cast of gunmetal in one piece with the impeller, or formed separately of forged bronze and keyed to it. This spindle runs in *lignum-vitæ* bearings, which are lubricated with water. The casing is formed in two parts to enable the impeller to be inserted, and the joint should, if possible, be so arranged that the impeller may be examined without disconnecting either the inlet or discharge pipes. In some successful pumps by Gwynne the side plates are entirely omitted.

The circulating pumps take their suction from a large screw-down inlet valve on the bottom of the ship. The discharge is through similar valves on the ship's side.

Under-water valves and fittings.—All holes in the hull of a ship below the water-line for the supply or discharge of condensing water or any other purpose require to be fitted with valves. The old wooden and composite ships were almost always fitted with *Kingston valves*, which are simply conical valves opening outwards, so that the pressure of the water tends to keep them closed. The valves are fitted with long spindles, which are brought inside the ship through stuffing-boxes, to enable the valves to be worked from inboard. They were, however,

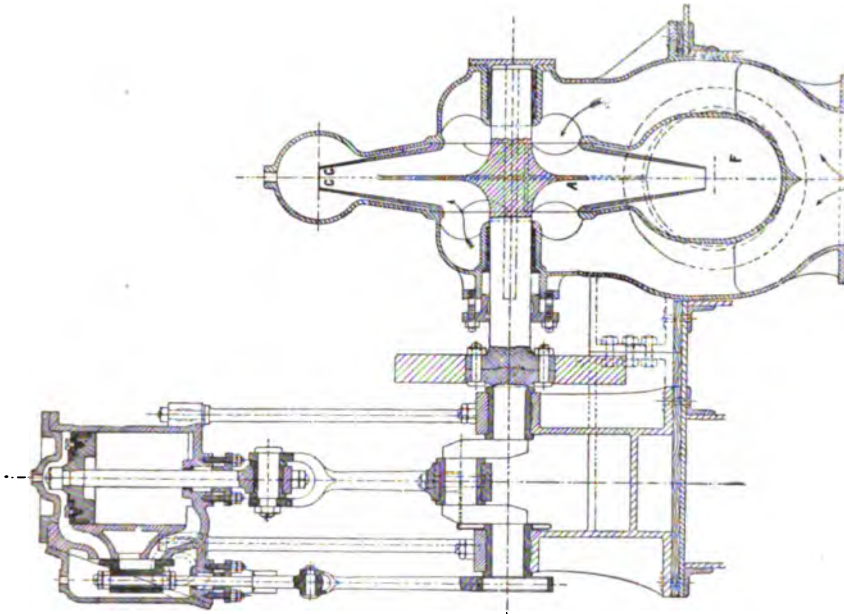


FIG. 240.

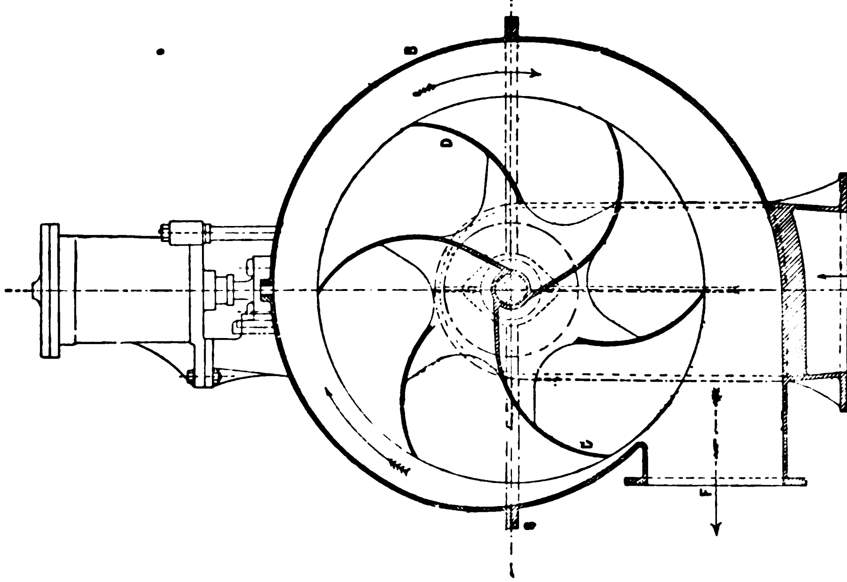


FIG. 239.

particularly suitable for wood ships, because they enabled a firm and secure connection to the hull to be conveniently made, as shown in Fig. 241, but were also often fitted to steel vessels. For iron or steel hulls, however, they have no special advantage, and in modern ships the cheaper and lighter ordinary screw-down valves are generally fitted for all the under-water orifices, while another screw-down valve is usually fitted inside the ship in most of the pipes, either at the sea valve or not far from it, for additional security. The Kingston valve is still sometimes fitted for blow-out purposes.

The spindles of all under-water valves in the Royal Navy have to pass a tensile test, equal to half a ton per square inch of area of the valve; with this limit, however, that the maximum test load is not to exceed twelve tons whatever may be the diameter of the valve.

A sketch of an ordinary sea valve for a steel vessel with a double bottom is given in Fig. 243, which represents the inlet valve for supplying a circulating pump, the valve being fitted at the inner end of a tube between the two bottoms. Gratings of large area are fitted to all sea inlet valves to prevent entry of weeds and other foreign matter. A plan of the grating of Fig. 243 is shown in Fig. 242. The tubes must in sheathed ships all be made of gunmetal to prevent galvanic action, as zinc protectors are then useless. Fig. 244 shows the attachment of such a tube to a sheathed ship, with the means of preventing access of water to the steel outer bottom. In the case of blow-out valves where sudden variations of temperature occur when blowing out, the tube is fixed to the outer bottom, but passes through the inner bottom by means of a stuffing-box, and is not rigidly secured to it, thus allowing for expansion. This is shown in Fig. 245; a guard is fitted below the valve to prevent it being lowered too far. The tubes of steel bottom ships are made of steel, where they are large enough in diameter to be properly cleaned and painted, but the smaller ones in which this cannot be done are made of gunmetal. The under-water valve is attached by studs to a thick facing ring, secured for this purpose to the end of the tube, and a spigot is fitted on the valve casing, which enters the tube and protects the end of the tube from wasting away. All gunmetal under-water valves of iron or steel ships are fitted with zinc protecting rings immediately below the valve box or tube and well attached to the steel, to prevent the decay of the hull of the vessel by galvanic action in the neighbourhood of the valve. These zinc protectors are shown in the sketches.

Duplicate centrifugal pumps.—The usual practice in large ships is to fit, for each set of main engines, two centrifugal pumps and engines, each large enough to circulate all the water required for full-power working. This provides for the case of accident to any circulating engine or pump, and doubles the pumping power in the event of a leak. Each pump is fitted with separate sea and bilge suctions, and the valves for changing from sea to bilge suction are arranged to be easily accessible. In some vessels, where weight and space are of importance, one pump only is fitted for each set, the discharge pipes being connected across, so that either pump may, if required, supply both condensers.

Bilge suctions to circulating pumps.—The circulating pump suction pipe leading to the engine-room bilge is provided with a non-return valve

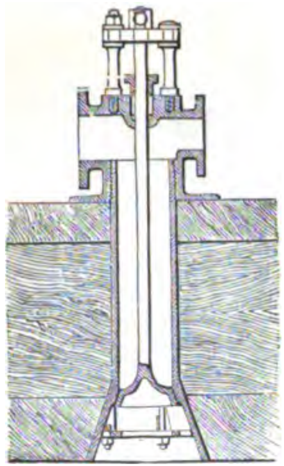


FIG. 241.

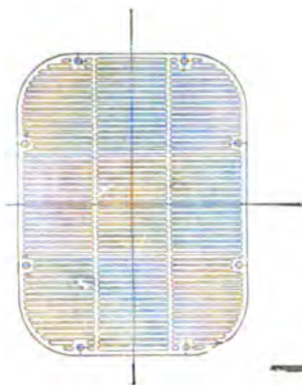


FIG. 242.

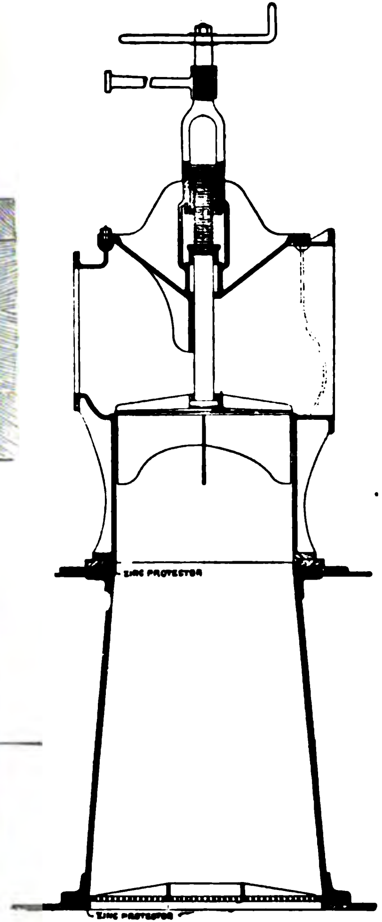


FIG. 243.

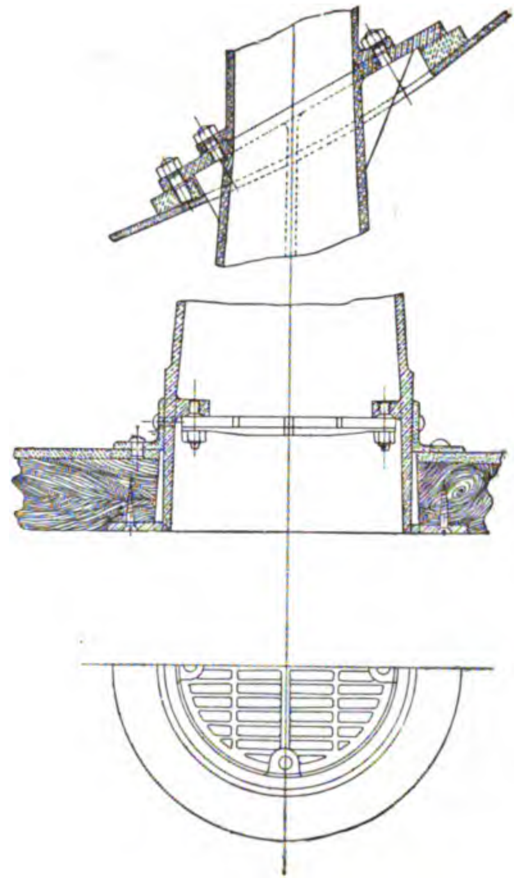


FIG. 244.

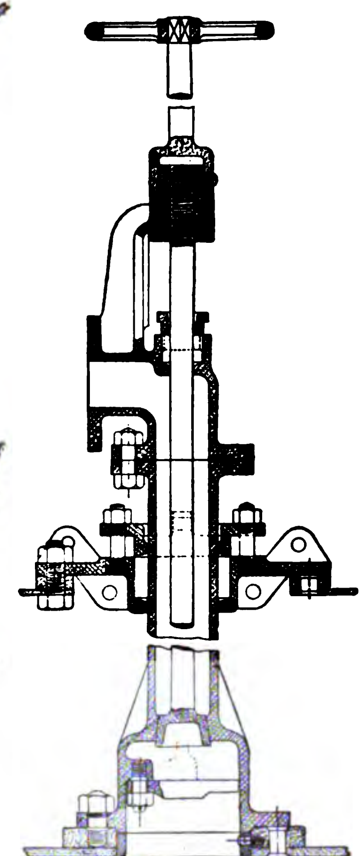


FIG. 245.

and strainer at its end, so that, in case of a serious leak, water might be drawn from the bilge and discharged overboard. These centrifugal pumps constitute by far the most powerful pumping appliances on board ship, the amount capable of being pumped out varying with the size of the vessel. In the largest battleships and cruisers, each of the four pumps fitted is capable of discharging about 1,200 tons of water per hour. Figs. 246 and 247 show the arrangement of pipes and valves usually fitted for this service, from which the procedure necessary to alter the pump suction from sea to bilge will be seen.

In this arrangement there is a common screw-down sea suction valve for the two pumps, B B being sluice valves fitted in the sea suction pipe, one to each pump. C C are the two bilge suction screw-down

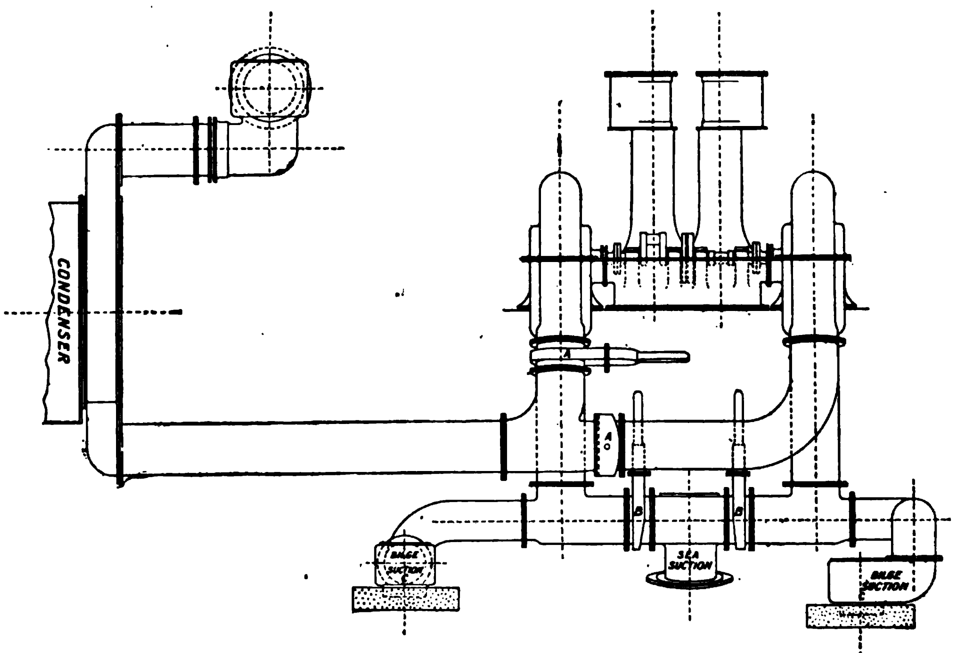


FIG. 246.

non-return valves, and A A are sluice valves fitted in the discharge pipes, one to each pipe. The pumps are not coupled together, but work quite independently, and it will be seen that to enable either pump to draw from the bilge its valve B must be shut, and the bilge suction valve C and the sluice valve A opened. Either bilge suction valve can be examined when the other pump is at work. The valves A and B are closed when it is required to shut off either pump when not at work.

Independent feed and bilge engines.—In many mercantile vessels and all ships of the Royal Navy the feed and bilge pumping engines are fitted as separate engines instead of attaching them to the main engines. Feed pumps when worked by the main engines, especially

by high-speed engines, are somewhat spasmodic in their action, and the pressure in the feed-pipes fluctuates more than is desirable. With separate feed-engines, their speed is regulated independently of that of the main engines, and is governed solely by the requirements of the boiler, so that the pressure in the feed-pipes is more uniform. Also the speed can be regulated so that the pump always draws a full supply of water, so that the feed-water is supplied to the boilers practically free from air, and this is conducive to their durability.

The feed-engines, main and auxiliary, are usually fixed in the stokeholds, so that the person in charge has full command of the feeding of his boilers, which is an important feature in ships that are subdivided into separate watertight compartments. The main feed-pumps draw from the feed-tanks only, but duplicate sets of pumps are fitted which draw both from the feed-tanks, the reserve fresh-water tanks, and from the sea, thus providing for breakdown of any feed-pump.

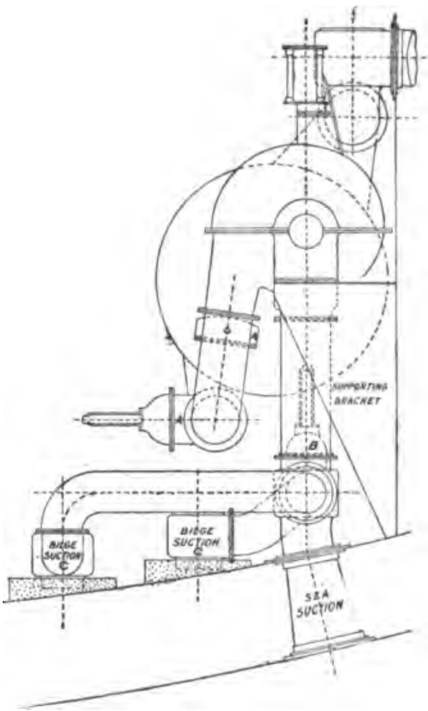


FIG. 247.

Vacuum gauge. — The vacuum in the condenser is indicated by a Bourdon gauge, similar in construction to Fig. 90, called the 'vacuum gauge.' This gauge is graduated in inches of mercury, and does not show, directly, the *absolute* pressure in the condenser, but only the difference between this absolute pressure, on the inside of the tube, and the absolute pressure of the atmosphere on the outside of the tube. As the actual pressure in the condenser is independent of the atmospheric pressure, the vacuum registered will, therefore, vary with the atmospheric pressure as recorded by the barometer. For example, suppose the constant condenser pressure to be four inches of mercury. Then, when the barometer shows 30·5, the vacuum gauge would register $30{\cdot}5-4=26{\cdot}5$; whilst if the barometer stood at 29·5 the vacuum gauge could only indicate $29{\cdot}5-4=25{\cdot}5$ inches, or one inch lower than in the former case.

The height of barometer should always be recorded when accurate data are required respecting condenser observations, and to enable comparisons of vacua to be made in various cases, the recorded vacuum should be corrected to the standard height of barometer of 30 inches, and the revised figure or 'corrected vacuum' enables records

on different occasions to be properly compared. If the barometer exceeds 30 inches, a corresponding amount is deducted from the vacuum gauge indication, and *vice versa*, to obtain the corrected vacuum.

Referring to an indicator diagram, Fig. 248, assume the actual pressure in the condenser to be the same in each case. If the barometer stand at 30·5, the distance of the zero line O P from the atmospheric line indicated by the line A A will represent a pressure due to 30·5 inches of mercury. If the barometer stand at 29·5, the atmospheric pressure will be less than before, so that the distance between the zero line, which remains in the same position, and the new atmospheric line indicated by the dotted line B B represents only a pressure due to 29·5 inches. The back pressure line C C being the same, it is evident that the lower the barometer is, the less will be the vacuum indicated on the diagram below the atmospheric line, and *vice versa*. This does not affect the area of the diagram, but only the indicated vacuum.

To determine the maximum attainable vacuum in the condenser, the condenser temperature must also be known, and the corresponding vapour pressure ascertained from tables in Chapter III. If this pressure be deducted from the atmospheric pressure given by the barometer, the remainder will be the maximum attainable vacuum with that temperature of condenser. The difference between the vacuum shown on the gauge attached to the condenser and the maximum attainable will be due to the slight pressure required to lift the air-pump foot valves and to air leaks or other inefficiency in the action of the condenser, air-pumps, &c.

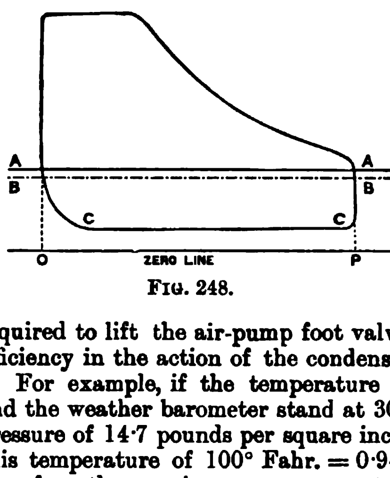


FIG. 248.

For example, if the temperature of the condenser be 100° Fahr., and the weather barometer stand at 30 inches, which is equivalent to a pressure of 14·7 pounds per square inch, the pressure of vapour due to this temperature of 100° Fahr. $= 0·942$ pounds per square inch, and therefore the maximum vacuum attainable in the air-pump will be $14·7 - 0·942 = 13·758$ pounds per square inch below the atmospheric pressure, which would be represented by about 28 inches on the gauge. The vacuum in the condenser, as shown by the gauge, must be slightly less than this, owing to the air-pump valves, but it may be still lower due to the causes mentioned above. If, for example, there is any air set free from the water, it will increase the pressure or reduce the vacuum although the temperature remains unaltered; hence the desirability of liberating all air from the feed-water before it enters the boiler, as in the feed-tank.

Heat rejected into condenser.—Suppose the terminal pressure in the cylinder to be 10 lbs. absolute and the back pressure to be $3\frac{1}{2}$ lbs., the temperature of the water after condensation being 100° F., which is a good working temperature for the condenser. The quantity of heat per lb. of steam rejected into the condenser is made up as follows :—

(*a*) The heat given up by the steam released from the cylinder, in condensing to water at 100° F. this quantity or that previously represented by I, see page 31.

(b) The work done on the steam by the piston in overcoming the back pressure.

The steam at the end of expansion will be assumed to be dry and saturated, although in practice it is more frequently wet.

The temperature corresponding to 10 lbs. absolute is 193·2° F., and the specific volume of steam at this pressure is 37·8 cubic feet, substituting these values

$$\begin{aligned} I &= t - t_0 + \cdot 966 - \cdot 7\,(t - 212) - \cdot 1851\,p\,V \\ &= 193\cdot 2 - 100 + 966 - \cdot 7\,(193\cdot 2 - 212) - \cdot 1851 \times 10 \times 37\cdot 8 \\ &= 1002\cdot 4, \end{aligned}$$

which is the amount of heat liberated by the steam in condensing to water at 100° F.

The work done by the steam in overcoming the back pressure is equal to $P_b V$ foot lbs. or $\cdot 1851\ p_b\ V$ British thermal units $= \cdot 1851 \times 3\cdot 5 \times 37\cdot 8 = 24\cdot 4$, so that the total heat rejected is $1002\cdot 4 + 24\cdot 4 = 1026\cdot 8$ thermal units per pound of steam, which is all abstracted by the condensing water.

It should be noticed that this number does not vary much whatever the final temperature of the water, as the latent heat part of the total is very great compared with the sensible heat part.

This quantity of about 1,030 thermal units is called the *heat rejected per pound of steam used.* To ascertain the total 'heat rejected' we must know the number of pounds of steam used per minute. Assuming this to be known, we get by multiplication the 'heat rejected' per minute. The heat imparted in the boiler to the steam will always be found to be greater than the heat rejected, and this represents the quantity of heat which is converted into work and is measured by the heat equivalent of the indicated horse-power.

If H = heat imparted in boiler per minute, R = heat rejected per minute, and I.H.P. = number of horse-power, then

$$H - R = \frac{\text{I.H.P.} \times 33{,}000}{778}$$

since one thermal unit is equal to 778 foot-pounds of work. In this statement the small loss by radiation is neglected.

Quantity of condensing water required.—We saw above that about 1,030 thermal units were rejected into the condenser per lb. of steam used. Now suppose in a surface condenser that the circulating water is raised in temperature to the extent of 20° to 25° Fahr. by passing through the condenser, then the number of pounds of circulating water per pound of steam condensed must be $\frac{1{,}030}{20 \text{ to } 25} = 51$ lbs. to 41 lbs. For vessels whose service may take them into the Tropics, where the temperature of sea-water in the summer is often 85° Fahr., it is not usual to allow for a greater rise of temperature than 20°, although in colder climates a smaller amount of water with a greater rise in temperature will suffice.

Suppose, next, we are dealing with a jet-condenser, and that the temperature of the injection water is 60° Fahr. In this case the injection water is raised by mixing with the steam to the 100° Fahr.—i.e. a rise of 40° Fahr., so that the quantity of injection water required would be $\frac{1{,}030}{40}$ = about 26 lbs. per pound of steam. The higher the

temperature of the injection water the greater will be the quantity required. As the feed-water for boilers supplied from jet-injection condensers is practically as salt as sea-water itself, frequent blowing out of a portion of the water in the boilers was necessary to prevent undue incrustation on the heating surfaces, and resulted in a considerable waste of heat.

We will now give some calculations respecting the operation of blowing out, which was of great importance in the old jet-condensing days, but is of minor importance now. The principles should, however, be well understood, as they are necessary when investigating the action of evaporators, and even with modern boilers, owing to occasional leakages of the main condenser tubes, the feed-water may have a proportion of sea-water mixed with it, and the effect of this should be considered.

The following calculations are general and will apply to any of the cases just referred to.

Quantity of water to be blown out to maintain constant density.—The proportion necessary to be blown out from an evaporator or boiler to keep the water at any particular density may be calculated thus :—

Let x = quantity of feed-water,
„ y = „ „ water to be blown out,
Then $x - y$ = „ evaporated.

Suppose the water in the boiler or evaporator to be kept at a density equal to n times that of the feed-water, then the quantity of solid matter blown out is proportional to $n \times y$. The quantity of solid matter pumped in with the feed-water during the same period is proportional to x, and since the density of the water in the boiler or evaporator remains constant, the solid matter blown out must be equal to that pumped in.

Therefore $x \times 1 = y \times n$ and $y = \frac{x}{n}$.

Case 1. If the density be kept at twice the density of the feed-water,

$$n = 2, \text{ and } y = \tfrac{1}{2}x.$$

In this case the quantity blown out must be half the total feed-water.

Case 2. Suppose the density in either a boiler working in connection with a jet condenser, or in an evaporator, to be kept at three times that of the feed-water, which in these cases would be sea-water, then

$$n = 3 \text{ and } y = \tfrac{1}{3}x,$$

i.e. the quantity necessary to be blown out to keep the density constant at three times the density of the feed-water is one-third the quantity of feed-water admitted.

Case 3. Suppose we have a modern engine with surface condenser, and that the tubes of the latter are leaking, so that the density of feed-water is 1 on the naval hydrometer—i.e. $\frac{1}{10}$ the density of sea-water—and that the density in the boilers is to be limited to four times that of sea-water ; in this case $n = 40$, and $\frac{1}{40}$ the total feed-water must be blown out.

If the machinery referred to is working at 500 I.H.P. and uses

15 lbs. of steam per I.H.P. per hour, the quantity of feed-water used per hour would be $500 \times 15 = 7{,}500$ lbs., so that 750 lbs. of sea-water enters the boilers per hour. The quantity of sea-water that must enter to raise the density to four times that of sea-water, is four times the weight of water the boilers contain. Knowing this latter weight, the time it will take to raise the density to any point can be determined by division.

The following calculations are principally given as being important in investigating the efficiency of evaporators. They are of little or no importance as regards boilers with modern machinery.

Heat wasted by blowing out.—

Let T_1 = temperature of the water in the boiler or evaporator;
„ T_2 = „ „ feed-water of the boiler, or inlet water to evaporator.

Each pound of water blown out has been raised in temperature from T_2 to T_1, so that the total amount of heat wasted by blowing out is

$$y\,(T_1 - T_2) \text{ thermal units.}$$

The total amount of heat expended on the x lbs. of water admitted to the boiler or evaporator consists of the quantity necessary to raise the whole x lbs. from T_2 to T_1, and also to evaporate $(x - y)$ lbs. at the temperature T_1, and is therefore equal to

$$x\,(T_1 - T_2) + (x - y)\,\{966 - \cdot 7\,(T_1 - 212)\}.$$

The proportion of heat wasted is therefore equal to

$$\frac{y\,(T_1 - T_2)}{x\,(T_1 - T_2) + (x - y)\,\{966 - \cdot 7\,(T_1 - 212)\}},$$

and since $x = n\,y$, the proportion wasted equals

$$\frac{(T_1 - T_2)}{n\,(T_1 - T_2) + (n - 1)\,\{966 - \cdot 7\,(T_1 - 212)\}},$$

We will now apply this formula to a few cases, selecting first an example of the jet-condenser, representing the practice of many years ago.

Suppose the working pressure in the boiler to have been 30 lbs. per square inch, the density to be kept at 20°, or twice that of sea-water. Temperature of boiler water, $T_1 = 270°$ Fahr., and of feed-water, $T_2 = 100°$ Fahr. Here $n = 2$, and by substitution we find that the proportion of heat wasted

$$= \cdot 1376 \text{ or } 13\tfrac{3}{4} \text{ per cent.}$$

Similarly, if the density be kept at 30°, $n = 3$, and the fraction of total heat wasted $= \cdot 074$, or about $7\tfrac{1}{2}$ per cent. We see, therefore, that the higher the density was kept, the less was the loss of heat by blowing out.

Take next the case of an evaporator receiving sea-water at 50° Fahr., and evaporating it at a pressure of 40 lbs. per square inch, the water being kept at a density equal to three times sea-water by brining. By substitution the loss of heat by the necessary blowing out is 9·3 per cent.

If the formula be applied to Case 3 of the preceding page, it will be found that the waste of heat would be very small should the boilers be worked under these circumstances.

CHAPTER XXI.

ROTARY MOTION.

Crank and connecting rod.—The mechanism used for the transformation of the reciprocating motion of the piston into the rotary motion of the shafting and propeller consists of the connecting rod and crank-shaft, the motion of which may be readily understood by reference to the outline diagrams, Figs. 249 and 250, the direction of rotation being the same in each, as indicated by the circular arrows.

The rotating shaft S, which is carried in suitable bearings, has on it a crank or arm S C, connected to the piston-rod by the connecting rod C B, which has a working joint at each end. The end, B, of the piston-rod is constrained to move in a straight line by the action of a suitable guide G G. It is thus easily seen that the reciprocating motion of the piston is transformed, through the medium of the connecting rod B C, into the rotary motion of the crank-shaft S, from which the motion is communicated to the propeller. The force P acting through the piston-rod, and which we will assume to be constant, is opposed by the resistance offered to the revolution of the crank by the action of the propeller. This produces either thrust or tension in the connecting rod, according as the crank is being pushed or pulled round. The arrows indicate the directions of the forces acting upon the joint B. The resultant of P and Q, as will be readily seen on reference to the diagrams, is always a force pressing on the guide G G for the direction of rotation shown, and is balanced by the equal reaction R of the guide which forms the third force acting on the joint B.

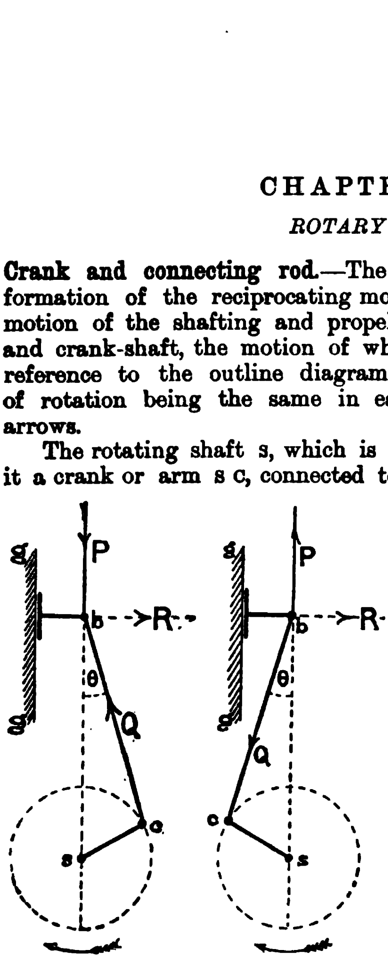


FIG. 249. FIG. 250.

Forces acting.—By applying the principle of the parallelogram of forces, it will appear that for any angle θ the connecting rod makes with the line of motion we have

$$P = Q \cos \theta \quad (1)$$
$$R = Q \sin \theta \quad (2)$$

also from (1) $Q = P \sec \theta$, and substituting the value in (2), then $R = P \tan \theta$, also $Q = \sqrt{P^2 + R^2}$.

When the crank is at the dead point the force Q on the connecting rod is obviously equal to the force P, and the pressure R on the guide vanishes, as may be shown by putting $\theta = 0$ in above relations.

Direction of rotation.—The direction of rotation shown in Figs. 249 and 250, which causes the piston-rod head to be always pressed against the guide, is that adopted for ahead working.

If the direction of rotation be reversed, the resultant of the forces P and Q will no longer press on the guide G G, but will act in the opposite direction, and an opposing guide surface will be required to balance this. The guide surface for 'ahead' motion is generally made larger than that for 'astern' motion, as engines rarely have to work astern at full power for any length of time.

In the horizontal trunk engines, which had no piston-rod and no other guide than the cylinder barrel itself, the direction of ahead motion was made the reverse of that indicated above, so that the resultant of the forces P and Q was an upward force, and tended to lift the weight of the piston and trunk, and prevent excessive wear of the cylinder.

No loss involved by using a crank.—It is important to notice that no power is lost by the intervention of the crank, for the force transmitted by the connecting rod to the crank pin can be resolved into two parts: one along the crank, and the other at right angles to it. No motion of the crank pin takes place in the direction of the crank, so that this component does no work. The only work done is by the component at right angles to the crank, which is all usefully employed in rotating it.

It is also necessary to guard against the error of supposing that work is lost in consequence of bringing the masses of the pistons, rods, &c., to rest, and starting them again in motion in the opposite direction twice in every revolution of the engine, and that there is a resultant loss of efficiency in reciprocating engines. Although the pistons, rods, &c., during the first half stroke receive acceleration from the steam pressure on the piston, the work thus accumulated is given out by pressure on the crank pin during the retardation, so that nothing is lost, but the distribution only of the work is altered (see page 485).

Connecting rod.—Figs. 251 and 252 show the form of top and bottom ends most generally employed for connecting rods. On Fig. 251 the connecting rod has a T-shaped bottom end and the brass is spigoted into it; the top end is forked, with brasses to fit on 'outside' gudgeon pins. In Fig. 252 a 'solid' head is shown, where the brass fits into a semicircular recess cut out from the end of the connecting rod; the top end here is also forked, but is shown with a gudgeon pin between the jaws, which works in brasses in the end of the piston-rod.

The 'solid' head is on the whole the most efficient, especially for large engines, and it is usually adopted in marine engines.

The brasses of connecting rods do not butt on each other, but have distance pieces between them, so that as they wear the liners can be taken out and thinned to allow of adjustment, or removed entirely if sufficient wear has taken place. Thin sheet brass or tin plate liners, in addition to the thick cast distance piece or liner,

are generally fitted. To insure correct working and to prevent straining and bending of the bolts, it is important that the nuts when screwed up as necessary to give the requisite adjustment should cause the brasses to firmly grip the liners. The bolts of connecting rods are subject to considerable shock at the bottom end of the stroke when the tension comes suddenly on them at the

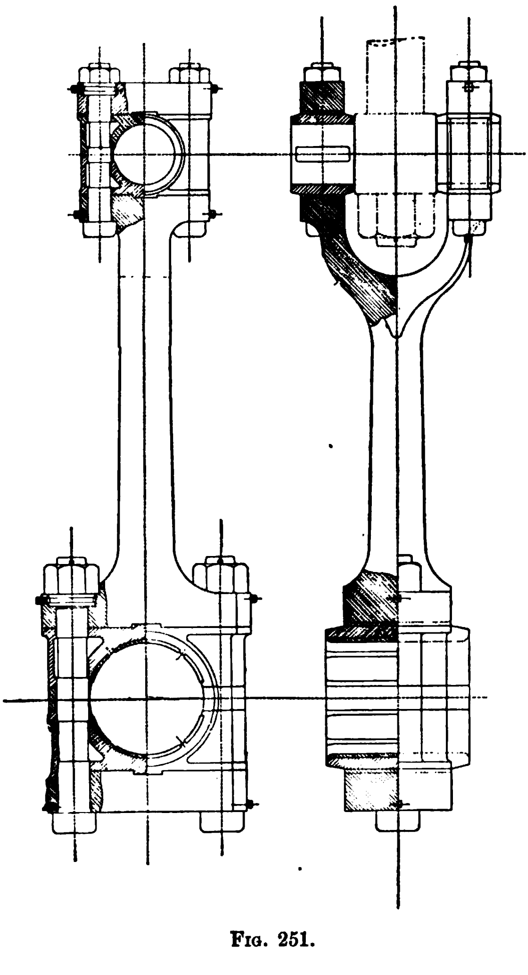


FIG. 251.

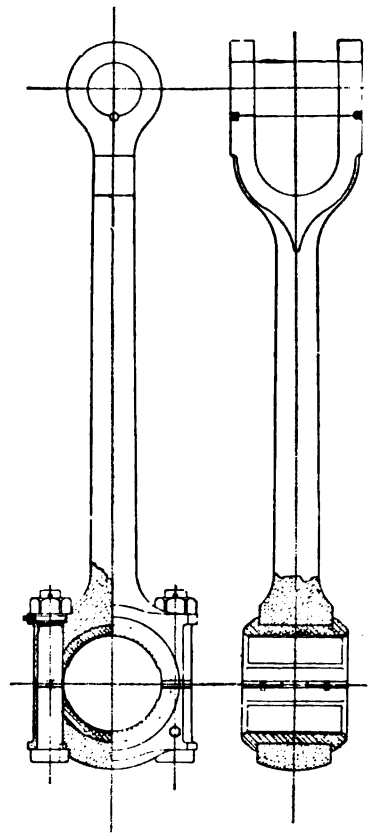


FIG. 252.

beginning of the up stroke, and this is much increased if there be any slackness in the bearings. There is no force acting on the bolts during the down stroke. They are, therefore, once in each revolution, alternately stretched by the steam pressure, and returned to their normal condition. This repeated stretching is, unless suitable arrangements are made concentrated on the smallest section below the nut—i.e. the sections at the bottom of the thread which are of small length, so that the bolts tend to break at this point if the stress is high. To prevent

this the bottom of the thread should be well rounded, and the weakest section is lengthened so that the stretching is not concentrated at one point, but over a considerable length of bolt, which reduces liability to fracture. The methods of effecting this are shown in Figs. 253 to 255, the area at the reduced section being made equal to that at the bottom of the thread, while the bearing surfaces are of the full diameter. Fig. 253 is inferior, as there is danger of the hole being drilled too far, and weakening the bolt. The same construction is adopted for main bearing bolts.

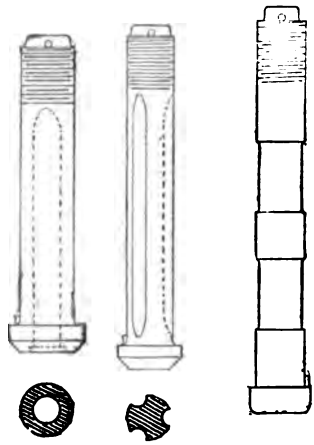


FIG. 253. FIG. 254. FIG. 255.

The nuts on the connecting-rod bolts are secured by means of set-screws, to prevent their slacking back when the engines are at work, and the bolts themselves are secured from turning by stops fitted under the heads, and from falling out when being disconnected, by set-screws screwed into them through the connecting-rod end, as shown in the sketches.

Crank-shafts.—Large crank-shafts are difficult forgings to make satisfactorily, and they are often made in separate pieces, having one crank on each, and joined by flanged couplings, or often, in the case of four-crank engines, a separate part for each pair of cranks. This simplifies the operations of forging and turning, and the several pieces are usually made symmetrical, so as to be interchangeable in case of accident. Fig. 256 represents the three-throw crank-shaft of a naval engine. The crank-shafts of smaller engines are forged in one piece. The various parts of the crank and other main shafting are, in the Navy, always filleted into one another, and are also made hollow, to obtain increased strength for the same weight of material; and the crank-shafts, arms, and pins are invariably made in one solid forging. The diameter of the hole is about 60 per cent. of the outside diameter.

In the mercantile marine, however, a cheaper construction is common, and gives satisfaction. This consists of 'built up' crank-shafts, the shafts, arms, and pins being separate forgings. This arrangement is shown in Fig. 257, which shows a three-throw crank-shaft in pieces. With this method the crank webs are shrunk on, and a pin is fitted to the shaft and driven in firmly, nearly to the full depth of the web, this pin being fitted part in the shaft and part in the web. A small groove is fitted in the pin to allow air to escape when driving. In some cases the crank-pin is entirely dependent on the shrinkage, but generally a screwed pin is fitted here, as indicated on the drawing. These pins are only shown on one of the pieces of shafting. It will be noticed that the mercantile marine crank-shaft is solid, and not hollow like the naval shaft.

Centrifugal lubricators.—The usual plan for lubricating the crank-pins of large engines when under way is shown in Fig. 258. The crank-pins being hollow, small holes are bored from the rubbing surface

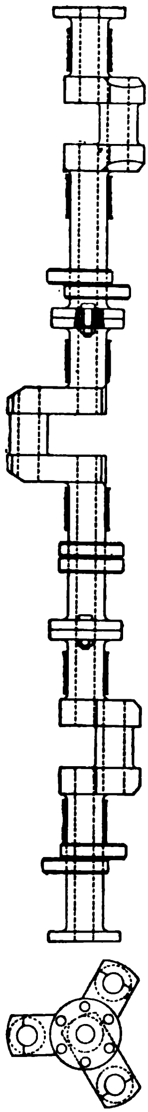


FIG. 256.

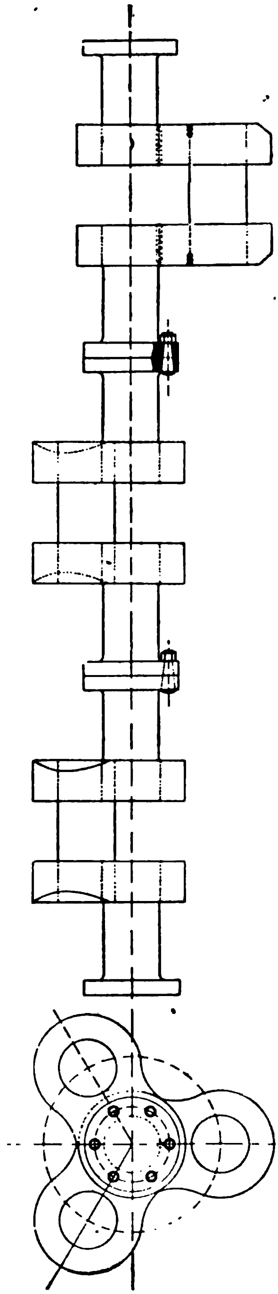


FIG. 257.

to the central space, as shown. An annular lubricator, A, on the crank-shaft is connected to the holes in the crank-pin, and by the revolution of the shaft the oil by centrifugal force flows as far from the centre of the shaft as it can, and makes its way through the small holes to the rubbing surface of the crank-pin. This system has been attended with satisfactory results, and is now applied to all high-speed engines. Many of the ahead eccentrics of the main engines are also similarly fitted with centrifugal lubricators, and the system is also extended to many of the auxiliary engines, which have to work for long periods (see Fig. 240).

Balance weights. — Balance weights were, in horizontal engines, usually fitted on the cranks opposite the crank-pin in order to counterbalance the weights of the crank-arms and connecting-rod heads, but later on this was found practically unnecessary, sufficient uniformity of motion in ordinary practice being obtained without these fittings, while in ordinary vertical engines the sum of the weights of the piston, piston-rod, and connecting rod, which reciprocate, is much greater than any balance weights that could reasonably be fitted. Balance weights are not now fitted except in special cases and on very fast running engines.

In the light quick-running engines of great power fitted in vessels such as torpedo boats and torpedo-boat destroyers, balance weights are often found to be required.

They are then, however, not fitted with the object of securing uniformity of turning moment on the shaft, but to counteract the forces arising out of the action of the engine, which cause a bending-moment to be

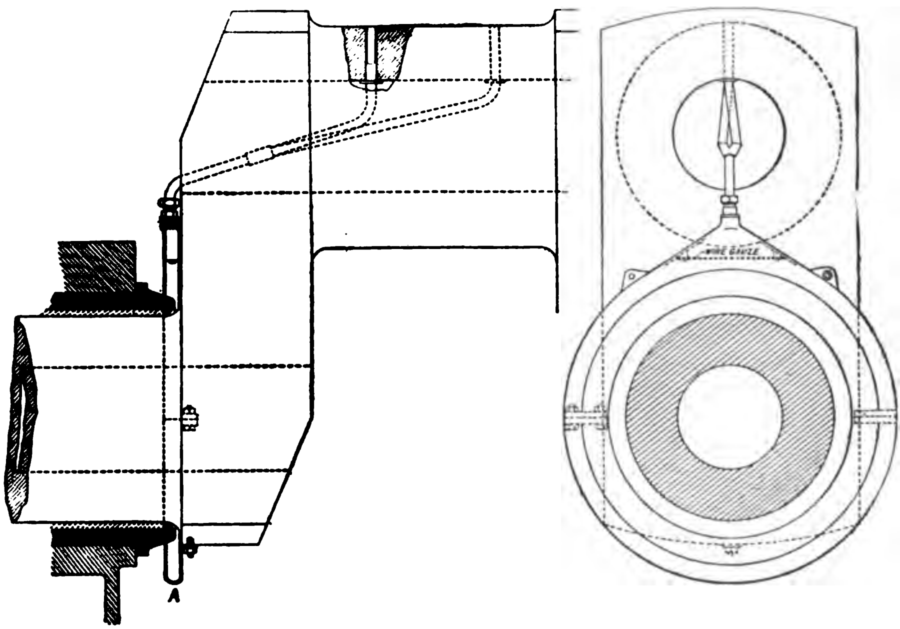


FIG. 258.

exerted on the vessel, and which, repeated with each revolution of the shaft, often cause excessive vibration in the hulls of such light vessels.

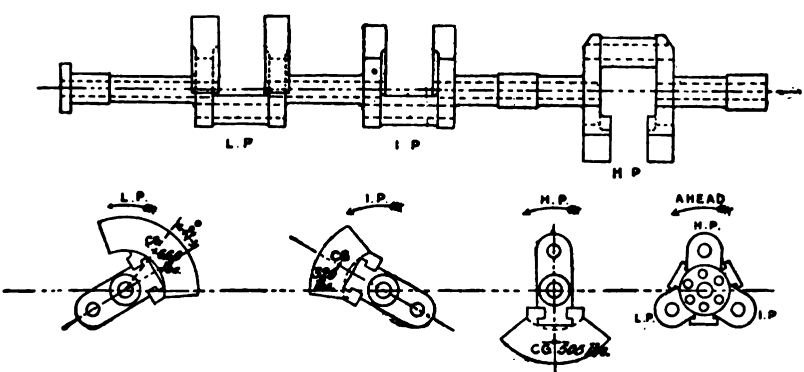


FIG. 259.

It is found that moderate balance weights placed in suitable positions, not necessarily opposite the crank-pins, are very effective in reducing vibration. Their size and position may be approximately

calculated in cases where they are proved to be required, but are generally ascertained by experiment by disconnecting the propeller shaft from the engine and running the latter at various speeds with different arrangements of weights. One such arrangement of balance weights fitted in 'Janus,' exerted a powerful effect in reducing vibration, and is shown in Fig. 259.

Turning wheel and gear.—On the after end of the crank-shaft a large worm-wheel is keyed, which is fitted for the purpose of enabling the

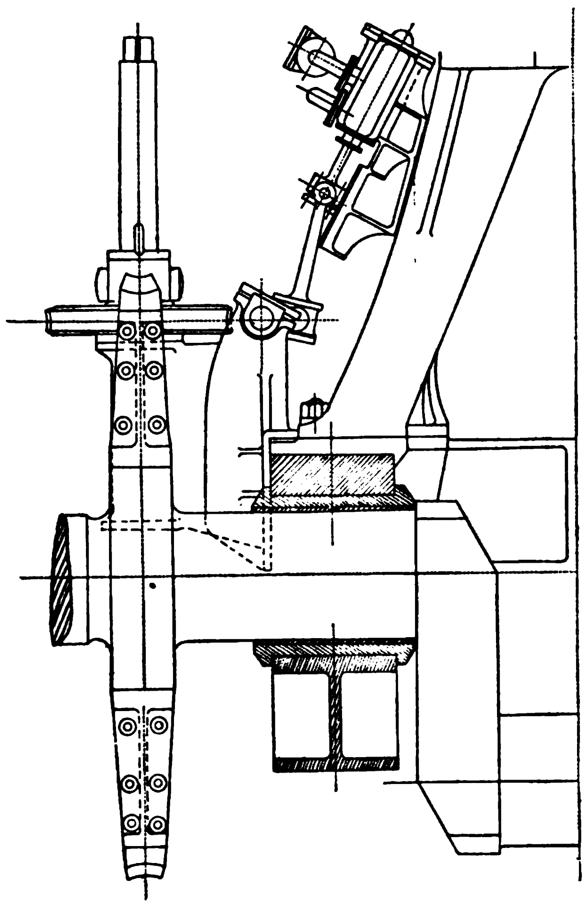


FIG. 260.

engines to be turned when not under steam. The worm is generally worked by a ratchet and lever in small engines. In large engines a small auxiliary engine is fitted to work the worm, so that the engines may be turned more rapidly, when under repair or adjustment.

This gear is so fitted that when steam is not available the worm can also be worked through the ratchet and lever by hand, suitable

disconnecting gear being fitted to the engine for use under these circumstances. Arrangements are of course necessary for disconnecting the worm from the worm-wheel when the main engines are required to be used, and in many cases the worm is entirely removed. An arrangement of turning gear is shown in Figs. 260 and 261, the position of the worm when disconnected being indicated by dotted lines. In the

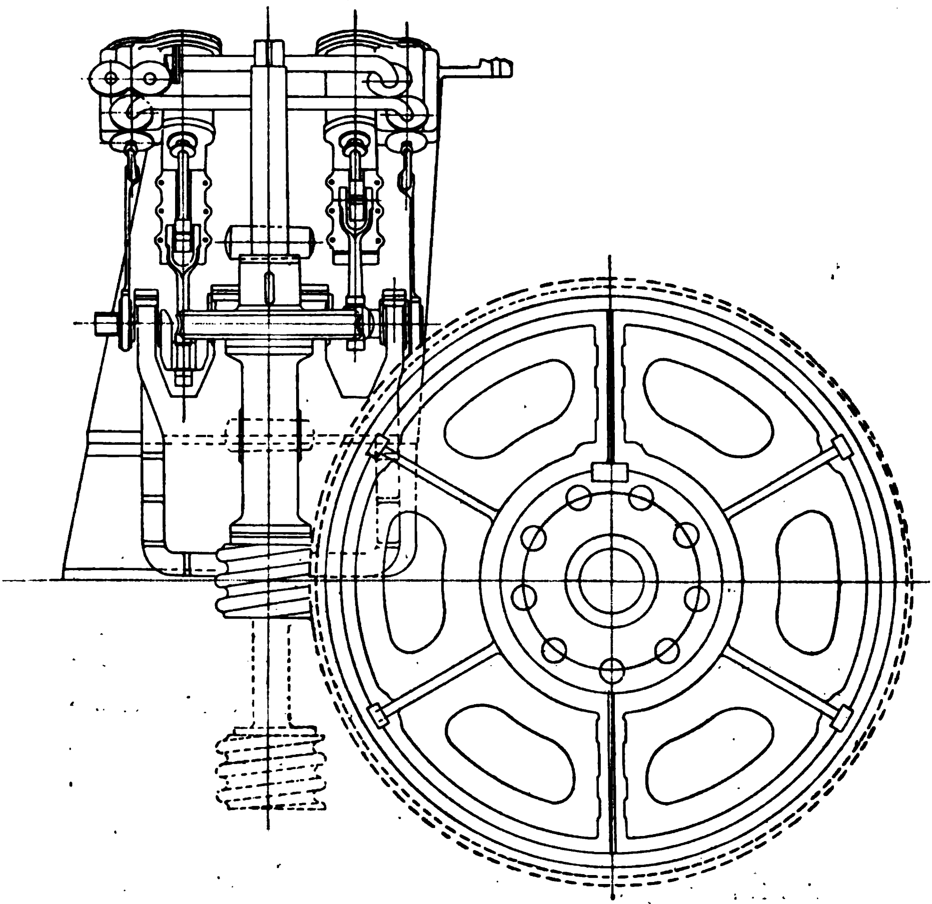


FIG. 261.

Navy it is required that the steam turning engines should be capable of turning the main engines through one complete revolution in eight minutes, when exhausting into the atmosphere. When not under steam the engines should be moved a little daily by the turning gear, to keep them in good order.

Main frames and bearings. Engine bearers.—The engine framing is a rigidly built up structure which holds the working parts in their correct relative position, provides the necessary guides, and the means

of attachment of the engine to the ship. In horizontal engines the main bearings generally form parts of strong cast-iron frames rigidly attached to the cylinders and the engine bearers. A sketch of one such engine frame is shown in Fig. 262, and the general arrangement of such an engine in Fig. 6.

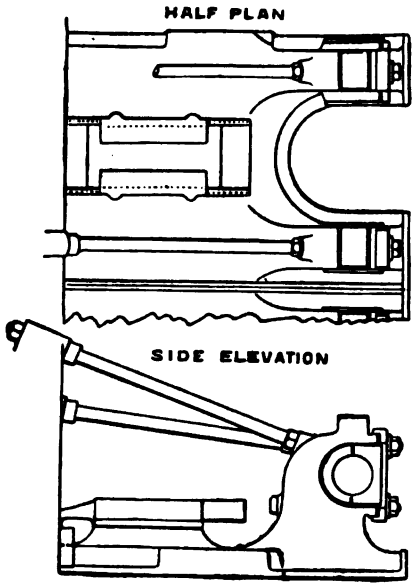


Fig. 262.

In vertical engines the crank-shaft bearings are contained in the *sole* or *foundation plate*, which forms the horizontal bottom part of the framework, and is rigidly secured to the hull of the ship by means of the *engine bearers*, which are a series of strong plate girders built up from the inner bottom of the ship. These engine bearers are generally of box construction, and their details will be found illustrated in Figs. 11 and 12. The sole-plate is in one casting in small engines, but in large engines it is constructed of a series of athwartship beams or girders carrying the main bearings of the crank-shaft, and fore and aft girders which rigidly unite these athwartship girders into one rigid structure. The cylinders are supported by standards or columns, bolted to the sole-plate. In some vertical engines the condenser forms one of the back standards, and is fixed on the sole-plate, wrought-iron or steel columns being employed for the front supports of the cylinders.

The framing now usually adopted for large engines consists of one of the following two plans :—

(*a*) A substantial cast-iron or steel column, fitted at the back of each cylinder, which carries the crosshead guides, together with two steel pillars at the front of each cylinder.

(*b*) Fitting four cast columns, two to each side of each cylinder.

Plan (*a*) is shown in Figs. 263 and 264. In this plan the crosshead guide is generally closed, as shown in the enlarged view, Fig. 265. The general arrangement of an engine with this form of guide is shown in Figs. 11 and 12. Plan (*b*) is shown in Figs. 266 and 267. In this plan there are two guides on each side of the cylinder, generally of cast-iron, and bolted to the cast columns as shown at G G in the sectional plan, Fig. 267, and the end of the piston-rod is attached to a crosshead with gudgeon pins at each side. These gudgeon bearings are attached to the slippers, which work in the guides, so that there are two gudgeon bearings to each cylinder. This plan is very efficient, although not so convenient of access for examination and repair as plan (*a*). It is, however, superior, in the fact that it provides

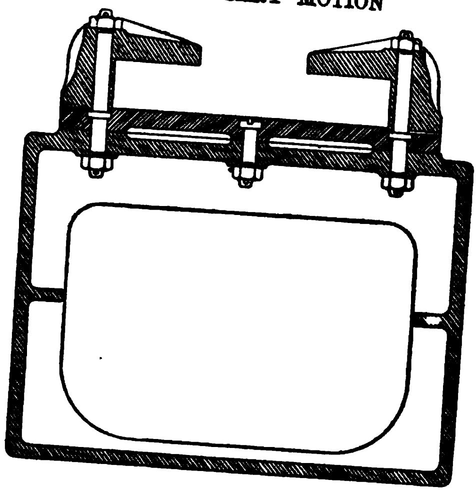


Fig. 265.

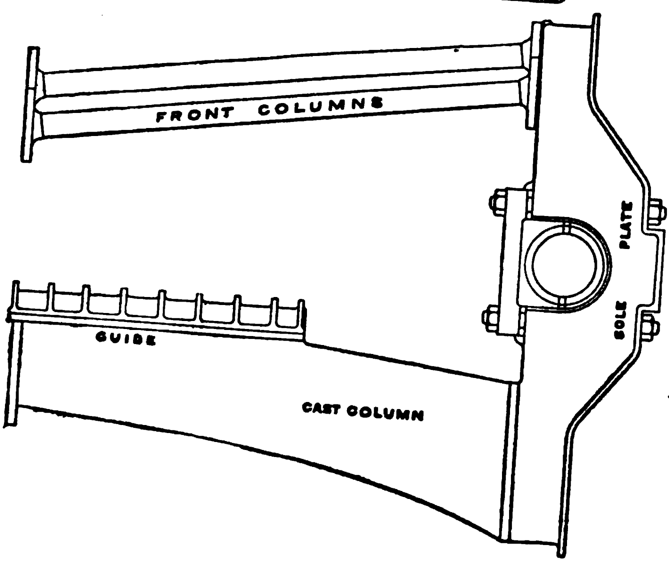


Fig. 264.

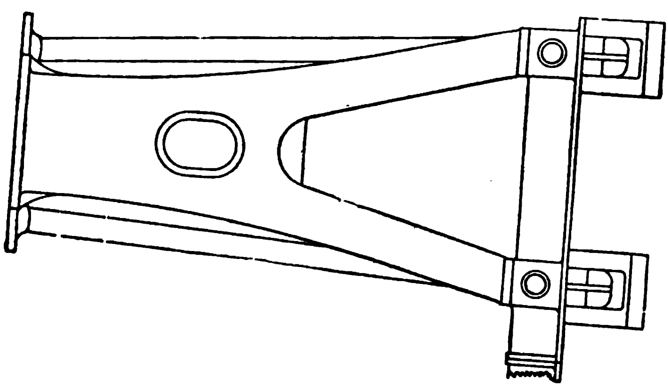


Fig. 263.

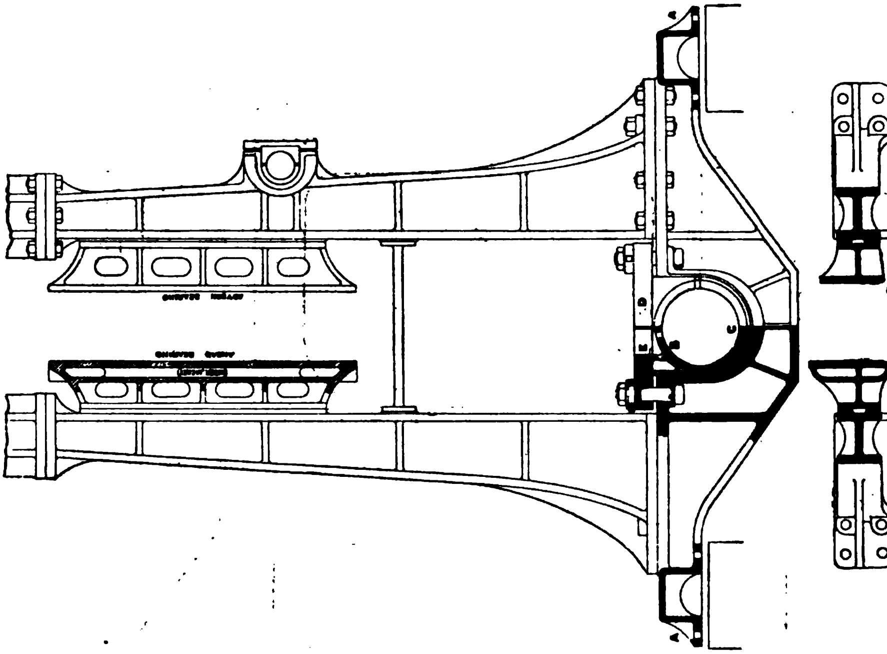


FIG. 267.

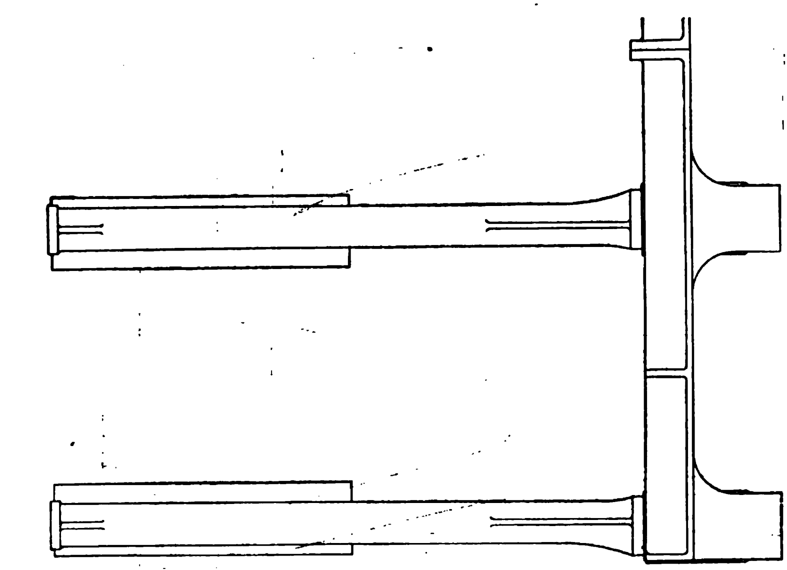


FIG. 266.

without difficulty the full area required for astern working, which, with plan (*a*), is more difficult.

In some engines, especially those of the mercantile marine, the front and back cast standards are identical and are one to each cylinder on each side, in which case there would be one gudgeon pin to each cylinder, and one guide at the centre of the cylinder. This plan is indicated by the sketches, Figs. 7 and 8.

In some large warships the framing has been entirely constructed of wrought-iron or steel suitably trussed to give sufficient rigidity, and fitted similarly to the framing of the small high-powered torpedo-boat destroyers described below. It has not often been adopted, however, for such large engines, as the extra expense is not compensated for by the small saving in weight which ensues.

The framing of very light high-powered marine engines, such as those of torpedo boats, torpedo-boat destroyers, and small cruisers, is generally formed of four steel columns attached to each cylinder and bolted at the lower end to the sole-platë and suitably stiffened by cross-stays. A sketch, sufficient to show the general construction of one such arrangement, is shown in Fig. 268.

Crosshead and guide.—The connection between the piston-rod and the connecting rod is made by means of a bearing termed the 'gudgeon pin bearing,' at which part is also fitted the crosshead bearing which works between the guides. There are two principal forms of these crosshead guides. In Figs. 269 and 270 are illustrated what is termed a 'closed' guide bearing which is suitable to the form of engine framing shown in Figs. 263 and 264. Sea-water is circulated at the back of the guide surface to assist in keeping it cool. The astern bearing surface is, as will be seen on the sketch, much less in area than the ahead bearing surface. In the other example (Fig. 271), called an 'open' guide bearing, and which is suitable for the type of framing shown in Fig. 267, the ahead and astern bearing surfaces are often identical. In all guide bearings of any size the ahead bearing surface is always lined with white metal, while in the closed variety the astern surface is now also generally so lined. White metal, which was at one time looked on with suspicion for gudgeon bearings, is now being fitted for such bearings in large engines, and is found to give satisfaction. The part which bears on the guide and carries the white metal, is removable for adjustment and repairs, and is called the *slipper*.

Main bearings.—The brasses in all main bearings should be so fitted as to allow the back or bottom brass to be removed for examination, without taking out the crank-shaft, and without removing the bolts. To effect this the back or bottom brasses are usually made concentric with the shaft, so that when the cap, and the outside or top brass are removed the back or bottom brass may be taken out by revolving the brass round the shaft. The crank-shaft journal revolves within two brasses, B, C (Fig. 267), fitted into each of the transverse sole-plates A A. As the principal forces in a vertical engine acting on the crank-shaft due to the piston are upward and downward, the brasses of the main bearings are placed at the top and bottom of the shaft. In horizontal engines they are placed at the front and back. The top brass B has a flat back on which bears a simple cap D, secured by

bolts and nuts, as shown, to the sole-plate A A. A handhole E in the cap enables the brass B to be felt while the engine is working, to ascertain whether heating of the bearing is taking place.

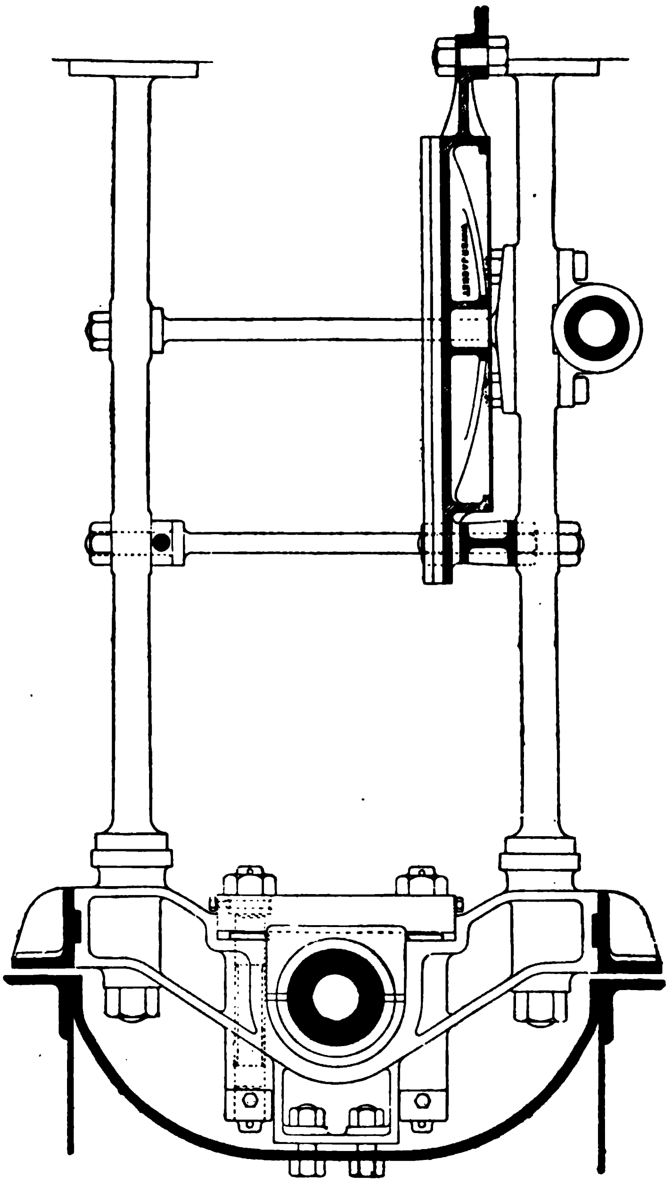
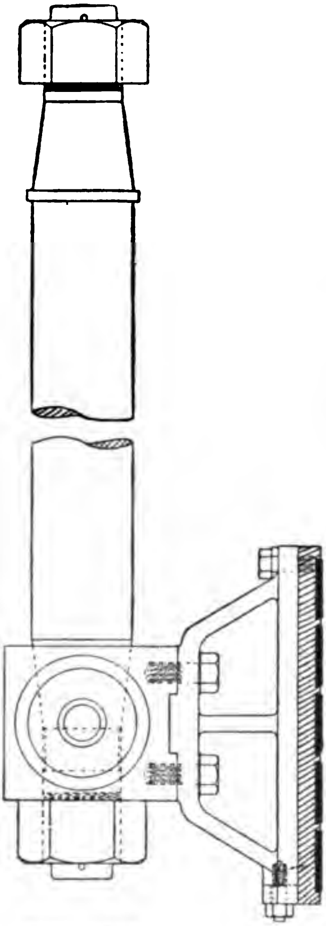


FIG. 268.

Fig. 269.



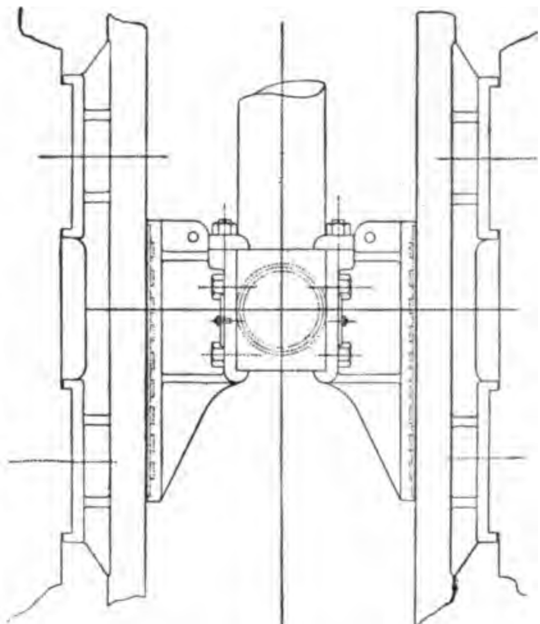


Fig. 271.

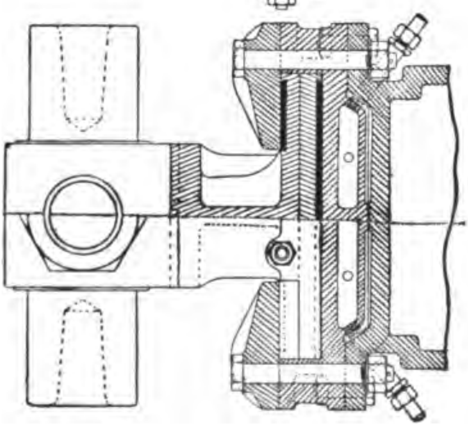


Fig. 270.

Water service.—The bearing surfaces should be so designed and arranged that, when properly adjusted, the ordinary lubricating arrangements may be sufficient to keep the journals, &c., from heating when the engines are being worked at full power. To provide, however, for the contingency of faulty adjustment, dirt getting into the bearings, or the friction being temporarily increased from any cause, small pipes with stop-cocks are led from one of the sea-valves to each of the principal bearings, to enable cold water to be run on them in case of their overheating. The crosshead guides on the ahead surface and also the thrust collars in recent ships are usually hollow to enable a stream of cold water to be circulated through them when under way.

White metal bearings.—The crank-shaft and crank-pin bearings and many other bearings in marine engines are filled with a soft white metal made of tin, antimony, lead, &c., with copper. The compositions used vary, but although there are numbers of cheaper patent compositions, Babbits' metal, an old mixture used for such purposes, still remains one of the best for ordinary work. This is composed of tin 10 parts, copper 1 part, and antimony 1 part, by weight.

These white metal alloys are soft and plastic, and if they be well lubricated will sustain a great pressure without heating. The white metal is confined by fillets or rims cast on the bearings, to prevent its being squeezed out, the depth of the recesses for the metal being usually about $\frac{3}{8}$-inch. In some cases the white metal is fitted as strips dovetailed into the bearings in the manner adopted for the lignum vitæ bearings for stern-fittings.

Propeller shafting. Plummer blocks.—From the crank-shaft the rotatory motion is communicated to the propeller by means of the screw-shafting, which consists of straight lengths of hollow forged steel shafting, carried in suitable bearings. The size of the screw-shafting between the crank-shaft and the stern tube, where the shafting leaves the vessel, is made smaller than the crank-shaft, since this portion has only to transmit the torsional stress due to turning the propellers, and not to stand any bending stresses such as are brought to bear on the crank-shaft. The bearings of this portion of the screw-shafting have therefore only to carry the weight of the shafting. These bearings are called 'plummer blocks,' they are usually made of cast-iron, and lined with white metal on the lower side to take the weight of the shaft. The upper part does not bear on the shaft. They are fitted with lubricating arrangements, and also a space into which water can be run in case the bearing gets warm. The water and oil boxes should be separate. Fig. 272 shows the general construction.

Shaft couplings.—The different lengths of crank and propeller shafting are secured together by means of ordinary flange couplings, as shown in Fig. 273. The flanges are forged in one with the shaft, and secured together by nuts and bolts which fit the holes in the flanges. The various lengths are, in the Navy, filleted into one another, which keeps them in line. For crank-shaft couplings the number of bolts used depends on the arrangement of the cranks. For instance, with a three-crank engine with cranks at equal angles, if the shaft is in three interchangeable parts, the number of bolts would be six or nine, so that the bolt holes would correspond when a piece of shaft was

shifted to a new position or a spare length fitted. Similarly, with cranks at right angles eight or twelve bolts would be used.

The aftermost coupling before the shaft leaves the vessel is a special one. The necessity for this arises from the fact that the length of

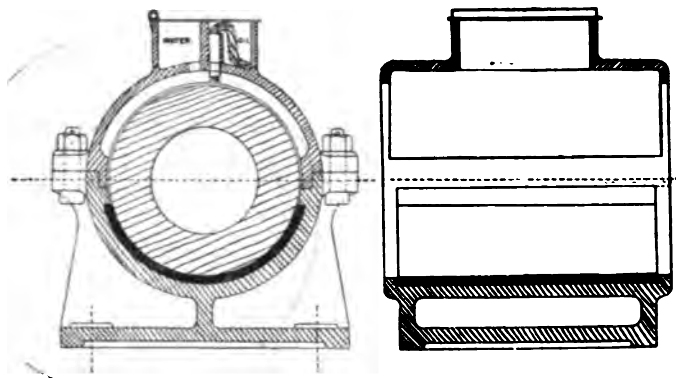


FIG. 272.

shafting in the stern tube has generally to be inserted in its position from outside the ship, owing either to the presence of a coupling at the after end of it, or for greater convenience in avoiding disturbance of internal parts of the vessel when withdrawing it for examination or

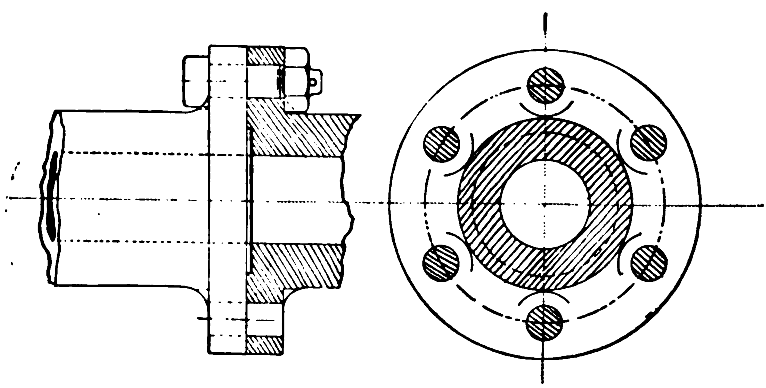


FIG. 273.

repair. To enable the shaft to be passed through the stern tube the forward coupling must therefore be separate and fitted after the shaft is in position. This coupling is often spoken of as a 'loose coupling,' and sketches of two forms of it are given in Figs. 274 and 275. It is found advantageous to make the coupling of wrought-iron if the shaft

is of steel. Steel couplings are often found, after a time to be so firmly adhering to the shaft as to necessitate cutting the coupling open to remove it. The turning moment is in each case transmitted by means of three or four keys recessed into the shaft and coupling. The difference in the two plans consists in the means of preventing the stern

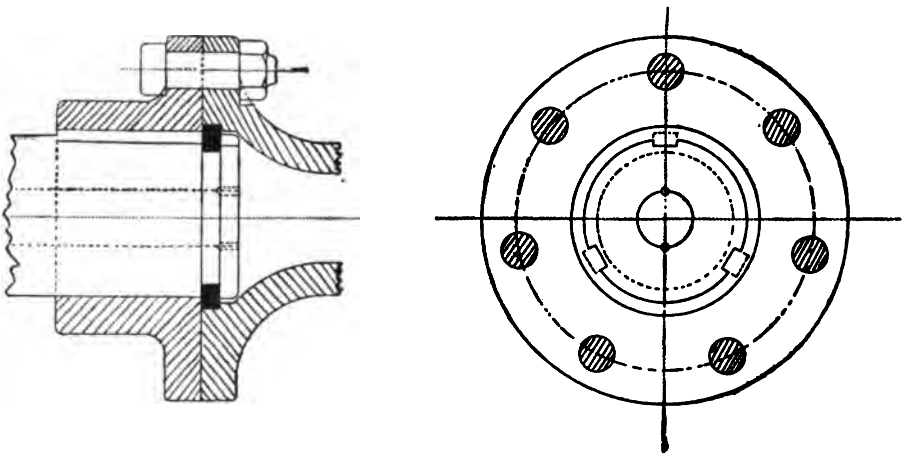


FIG. 274.

shaft from being withdrawn from the coupling when going astern. In Fig. 274 this consists of a recessed ring in halves, shown in the sketch in black. In Fig. 275 the end of the shaft is screwed, and a nut used; a pin being fitted to prevent the nut unscrewing. The nut or ring bears against the loose coupling and prevents the shaft from

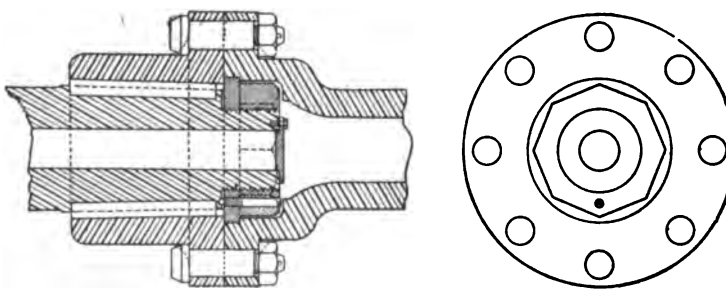


FIG. 275.

being drawn out. In each case a plate or plug is fitted to the hole in the stern shaft to prevent possibility of water passing into the vessel.

Thrust block.—On the foremost length of the propeller-shafting a bearing of special form is usually fitted, to receive the thrust of the propeller and transmit it to the ship. In order to reduce the intensity of the friction, the thrust journal on the shaft is made with several collars, which press on properly fitted thrust surfaces in the bearing, the

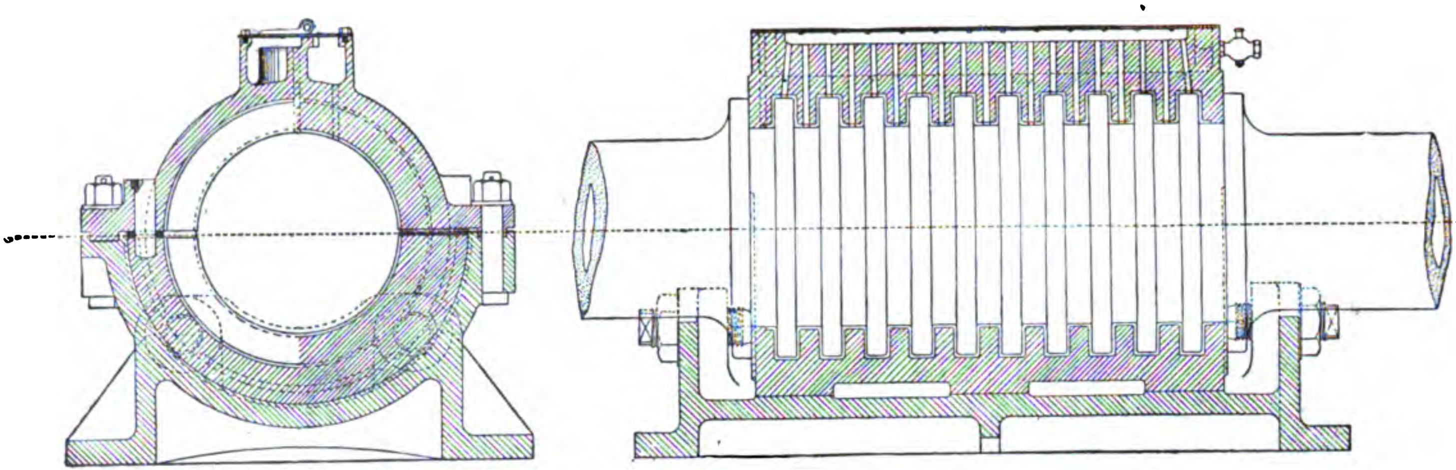


FIG. 276.

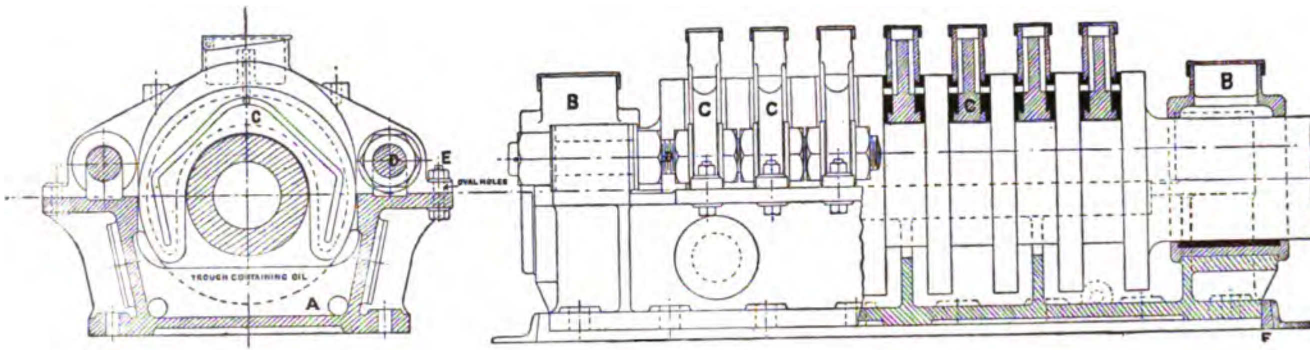


FIG. 276A.

pressure being exerted in one direction when going ahead, and in the reverse direction when going astern. Fig. 276 shows an arrangement which has often been adopted, in which the rubbing surfaces of the block are completely lined with white metal, and which arrangement has generally given satisfaction. The cap is filleted into the bottom half of the bearing, and oval holes are fitted in the base-plate to allow of adjustment in the fore and aft direction when wear takes place. This adjustment is necessary, otherwise the propeller would gradually wear its way forward and cause the cranks, eccentrics, &c., to be out of line with the corresponding parts of the engines, besides bringing side stresses on the crank-arms and bearings.

Another form of thrust block has often been fitted of similar general design, but containing separate brass thrust rings fitted in the bearing to form the rubbing surfaces. These are made in halves, and arranged so that they may be renewed when necessary. Both these plans of thrust block have, however, been generally superseded by a type which permits the position of each of the thrust rings to be adjusted independently by means of nuts and screws. This block, Fig. 276A, consists of a hollow trough A, which is kept supplied with lubricant, and ordinary plummer-block bearings B at each end. The shaft collars are formed as in the previous examples, but of larger diameter, and the parts against which the shaft collars bear consist of separate pieces shaped similar to horse-shoes, so that they can be removed and replaced without disturbing any other part. These 'horse-shoe' collars, C, fit between the collars on the thrust shaft, and are fixed in position or adjusted by nuts on two screwed bars, D, attached one on each side to the hollow trough foundation. The wedges F, sometimes fitted, also enable the block to be adjusted bodily. The horse-shoe collars are lined with white metal on each side, in which a groove is cut for the lubricating oil, and separate lubrication tubes are provided for the ahead and astern faces. Small bolts E are fitted to the collars, to prevent them rising from their proper positions.

Thrust blocks are carried on strong plate bearers generally fixed to not less than three frames of the ship, and wedges or screws are fitted to the foundation plates to enable the position of the thrust blocks to be adjusted bodily within certain limits if required. The holes in the base of the thrust block, for the holding-down bolts, are made elongated to admit of this. An ordinary plummer block should always be fitted close to the thrust bearing to carry the weight of the shaft, so that the thrust bearing will only have to sustain the thrust of the propeller and not take any of the weight of the shaft.

In single-screw ships the thrust of the propeller has sometimes been taken on a disc on the stern post fitted with lignum-vitæ segments. (See Chapter XXV.)

CHAPTER XXIA.

THE MARINE STEAM-TURBINE.

Rotary engines.—Having described the usual arrangements for producing the rotary motion of a shaft, which motion is essential for marine propulsion, we will consider an example of the 'rotary engine' which is being used to a considerable extent for this purpose. 'Rotary engines' are those in which the steam causes a rotating motion by direct action on the shaft, without the intervention of mechanism such as a crank and connecting rod.

Steam turbines.—The most successful and simple are those which act by the impulse of a mass of steam impinging on blades fixed on the revolving shaft exactly as in the turbine worked by water pressure. The steam may flow either in a radial direction or parallel to the axis, the latter variety being practically more efficient is described hereafter.

The principal difficulty in devising a satisfactory rotary steam-engine on this principle has been the excessive speed with which steam issues into a space containing a low pressure from an orifice under even moderate pressure, the behaviour of steam and water being quite different in this respect. The higher the velocity of the fluid, the higher must be the velocity of the blades on which it acts, if efficiency is to be secured, since theory and experience have shown that the peripheral velocity of the blades themselves must be about half the steam velocity through the blades. In the marine steam-turbine introduced and perfected by the Hon. C. A. Parsons, C.B., of Newcastle, this difficulty has been overcome by the device of splitting up the total fall of pressure into many stages, using in succession a large number of turbine blades, thus reducing the necessary speeds to practicable limits.

Another form of turbine, the Curtis, has recently been fitted for marine work in the United States and Japan. In this variety the number of stages of pressure-fall is much smaller than in the Parsons. There are also other forms of turbine, such as the De Laval, Zoelly Rateau, &c. The former is successfully used for land work and for electric lighting on board ship but not for marine propulsion. In the De Laval type no attempt is made to reduce the high speeds, and it still remains a single turbine variety, the single set of blades working at a much higher speed than the blades in other types of turbines, and the speed of the turbine shaft is reduced the required amount by helical gearing outside the turbine case.

The Parsons marine steam-turbine has been fitted in many vessels with success and economical efficiency. The first vessel to be so worked was a small one, the 'Turbinia,' since which a large number of mercantile vessels, including the largest and fastest afloat, and war-ships of all classes, have been so fitted. All war-ships ordered for the Royal

Navy since 1905 are fitted or being fitted with steam turbines, mostly of the Parsons type.

Parsons marine steam-turbine.—This consists of a comparatively thin hollow cylindrical drum supported at intervals by being riveted to the periphery of two or more wheels, the end wheels being supported by arms from bosses secured to the central shaft. This drum rotates on its axis, and is provided with a large number of inclined blades arranged in rings, and well secured in grooves in the revolving drum. There are a series of such rings of blades along the length of the drum, the complete arrangement being termed the 'rotor.' Between each row of revolving blades there is a corresponding ring of similar blades secured in the outer fixed cylinder or casing containing the revolving drum or rotor, but inclined at a different angle to that of the revolving blades, the moving and fixed blades being practically similar, but with angles reversed.

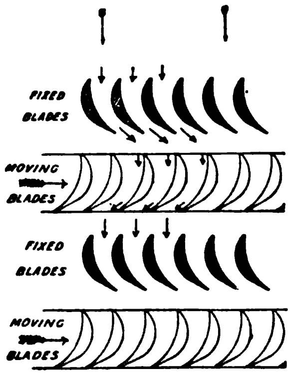


FIG. 276B.

Driving forces.—The steam leaves the fixed blades with a velocity having a high circumferential component, strikes the moving blades, then traverses the curved inner surface of these blades, leaving them with a relative velocity in the opposite direction. The rotating blades are therefore driven partly by impact and partly by reaction.

Fig. 276B shows the relative angles of fixed and revolving blades and the direction of motion of the revolving blades and of the steam. The angles and curvature of blades are so arranged that the velocity of the revolving blades and steam causes the latter to enter the moving blades with a velocity parallel to their surfaces; and similarly, on leaving the moving blades the combination of the velocity of these blades with the velocity of the steam along the inclined surface of the blade causes the final velocity of steam on exit from these blades to be parallel to the axis of the turbine.

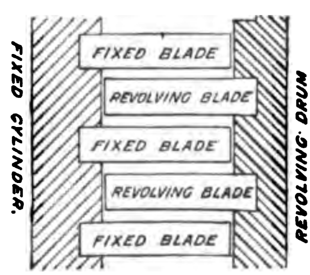


FIG. 276C.

The steam then proceeds through the next series of fixed blades, and so on till the condenser is reached. The small arrows in Fig. 276B indicate generally the direction of flow of the steam in its passage through the blades.

All the blades are fitted to the cylinder casing and rotor by being recessed in them as shown in Fig. 276C, and are secured at the correct distance apart circumferentially by packing pieces caulked between them in the recesses. It will be seen, therefore, that the angles of the blades cannot be altered, so that the turbine only works in one direction, and to secure astern working a separate turbine has to be fitted on the shaft with blades arranged in the reverse direction. When

going ahead the steam supply to the astern turbine is shut, and when required to go astern, steam is shut off from the ahead turbine and turned on to the astern one.

Theory of the water turbine.—The theory of a turbine working with steam, which changes its pressure volume and temperature in its passage through the turbine, is a complicated one, but the action is in many respects similar to that of a turbine working with an incompressible fluid such as water, and it will help to an understanding of the steam-turbine if the theory of a succession of water turbines be first considered, this being comparatively simple when friction, &c., is neglected, which we shall assume is done.

Consider first a single turbine. In Fig. 276D let C Y be the axis of the turbine, and C X the direction of motion of the moving turbine blades, also let A B represent the velocity in magnitude and direction of the jet of fluid impinging on the moving turbine blades, composed of a velocity A C along the axis of the turbine, combined with a velocity C B in the direction of motion of the blades. Bisect C B at D, then we know from hydraulics that the turbine works to best advantage when D B is the velocity of the moving blade, and the inlet angle of the turbine blade, or the angle of the revolving edge which receives the impinging fluid, is represented in direction by A D, and the magnitude of A D represents the velocity of the entering fluid in relation to the moving blade. Making C F = A C, then D F should be the angle of the moving blade when the fluid leaves it, and C F is the final velocity of the fluid leaving the turbine blade, i.e. in direction parallel to the axis of the turbine.

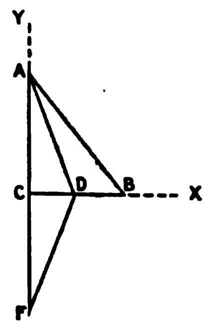


FIG. 276D.

The shape of the face of the turbine blade which receives the impinging fluid can now be determined, as lines drawn parallel to A D and D F are tangents to the curve at the inlet and outlet edges respectively, the intermediate portion being formed by a convenient smooth curve. The curve of the back part of the blade is formed from considerations of strength and to give the necessary area for the passage of the fluid.

As regards efficiency, we know that the energy in the fluid entering the moving turbine blade is proportional to AB^2, while the energy in it when it leaves the blade is proportional to CF^2. As $CF = AC$ the energy given up to the moving blade is proportionate to CB^2; the theoretical efficiency is therefore $\frac{CB^2}{AB^2} = \cos^2 ABC$.

The smaller the angle A B C is, the greater is the efficiency, and for a high efficiency this angle must be small, and consequently C B must practically be equal to A B, and in this case the speed of the turbine wheels must approximate to half the speed of the entering jet of fluid.

The velocity with which steam issues from a properly shaped orifice

under high pressure into a space such as a condenser, containing a very low pressure, is very considerable. With a pressure of 100 lbs. per square inch flowing through a well-proportioned divergent jet into a space containing a vacuum of 28 inches the velocity rises to about 4,200 feet per second, so that it will be seen that for the highest efficiency in a turbine with a single row of blades, the speed of the blades must approach 2,100 feet per second. Such speeds are enormous, and besides the practical difficulties of dealing with them, they lead to very extreme stresses in the revolving parts when the turbines are of any size. For this reason the highest speed adopted in large De Laval turbines is about 1,200 feet per second. For marine purposes it would not be possible to design an efficient propeller to work at such high speeds.

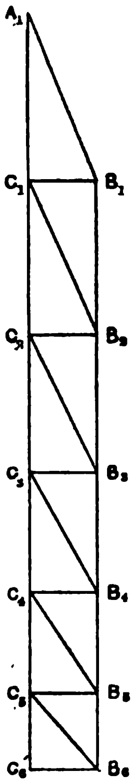


FIG. 276E.

We will now consider how the speed of rotation may be diminished, assuming as before that the fluid dealt with is water. Referring to the figure, it will be seen that if the speed C F at which the fluid leaves the turbine is allowed to increase, we can make the speed C D of the blades low, but then the amount of energy in the fluid leaving the blades is high, and the amount of energy taken up by the blades and hence the efficiency is low.

Let us assume that we make the speed of the turbine blades low, and now consider a series of elementary turbines, each fitted with a single row of blades. We can pass the fluid, after it has left the first turbine and containing a considerable amount of energy, through a second turbine, and get it to give up some more of its energy in the second turbine, and thence to a third and any number of turbines, each of which takes out some of the energy left, so that finally the steam leaves the last turbine with comparatively low velocity and containing but little energy.

By sufficiently increasing the number of turbines the speed of rotation can theoretically be reduced to any desired extent. The figure giving angles and velocities for a succession of such turbines is as follows: Let A_1B_1, in Fig. 276E, be the velocity and direction of the fluid impinging on the first moving blade, then half B_1C_1 is the velocity of the turbine blade, and we will assume this to be constant for the whole of the succeeding turbines, since they are generally arranged of about the same diameter.

The velocity of fluid leaving the first turbine is A_1C_1 and this turned by suitable fixed guide blades into the direction of C_1B_2 such that $C_1B_2 = A_1C_1$.

Draw B_2C_2 parallel to B_1C_1 then C_1C_2 is the velocity of fluid leaving the second turbine. This is turned by fixed guide blades into the direction C_2B_3 such that $C_2B_3 = C_2C_1$, and then impinges on the third turbine and so on. The velocity of fluid, leaving the six turbines for which the diagram is drawn, is reduced to C_5C_6, and the theoretical efficiency of the combination is $\dfrac{(A_1B_1)^2 - (C_5C_6)^2}{(A_1B_1)^2}$, so that if the final velocity C_5C_6 is small, the efficiency will be very high.

If V be the initial velocity of the steam, n the number of turbines; V_n the velocity of the steam leaving the last turbine, and $t =$ the turbine speed, it is easily deduced from Fig. 276E that $V^2 - V_n^2 = n4t^2$, hence $t^2 = \frac{V^2 - V_n^2}{4n}$, and t varies as $\frac{1}{\sqrt{n}}$ if the initial and final velocities are constant, so that if the number of turbine blades be increased fourfold, say, the speed of the turbines can be reduced by half and so on. In this way, in the Parsons' turbine, by increasing the number of blades, the velocity of the rotor is reduced to practical limits without sacrificing efficiency.

The preceding refers to a fluid such as water of constant density, but the action of steam in traversing the series of turbine blades is very much more complicated owing to its changes of density, temperature, &c., and the considerable space necessary for its full consideration cannot be afforded in a work of this kind.

In a ship the steam is passed through more than one complete turbine, and each main turbine, as hereafter described, generally works a separate shaft. As the steam passes through the turbine, the pressure falls, and consequently the volume increases, and in order to maintain a constant axial flow, as is usually desired, the volume between the blades and casing in the same turbine is generally increased from the inlet to the outlet end, this being done either by increasing the heights of the blades or by spacing them further apart, or by a combination of both. If the blade heights and the spaces between the blades were made to correspond exactly with the steam volumes, their tips would lie on a fair curve, that is, each row would be somewhat longer than the preceding one, but in practice this is not carried out, and the blades are arranged in sets of equal height as shown in Fig. 276L, and the spacing between the blades is varied.

Details of arrangement.—Figs. 276F and G show a vertical section through, and end elevation of, a high-pressure turbine. The glands and bearings are for clearness omitted from Fig. 276F, as they are shown in detail in later figures. The boiler steam, after passing through a strainer, enters the orifices A, Fig. 276G, proceeds thence to the hollow belt B, Fig. 276F, and then enters the turbine blades and proceeds through the fixed and revolving blades along the passage between the cylinder and rotor. At C an enlargement of diameter of cylinder occurs, the steam therefore expands and becomes of lower pressure. It then passes through the second series of fixed and revolving blades along the larger space between cylinder and drum until the point D is reached, where a further enlargement of cylinder diameter takes place with consequent further expansion of steam, still further expansions taking place at E, F, and G till the exhaust orifice H is reached. An end view and section of the forward rotor wheel or support is shown in Fig. 276H. In this case it will be seen that the wheel arms are hollow so as to admit steam to a space in the centre of the shaft, which, when steam is turned on quickly to the turbine, reduces the relative motion due to expansion, and so enables the initial dummy clearances allowed to be reduced and thus favours economy; this device also maintains as nearly as practicable the same dummy clearance under varying conditions of temperature.

With this system of steam propulsion there are usually either

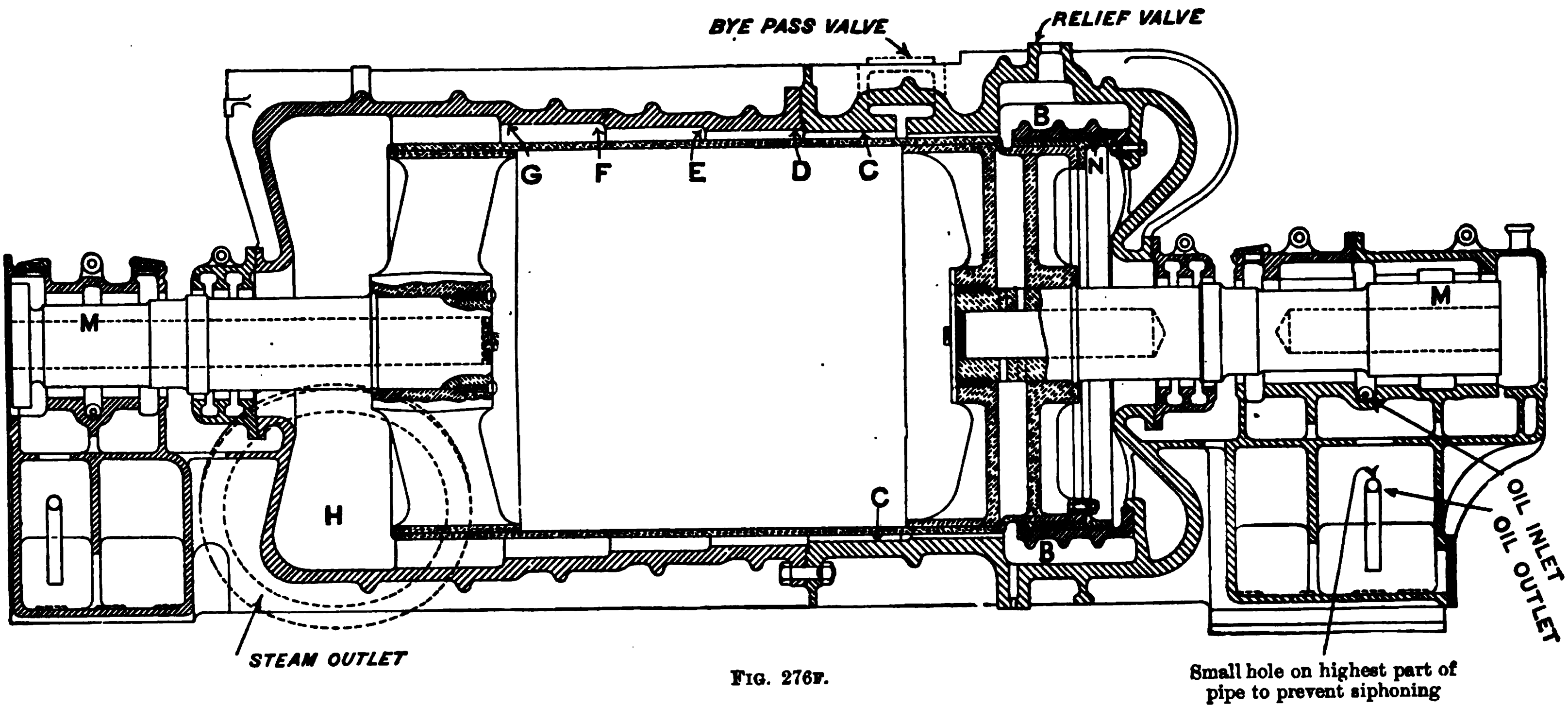

BYE PASS VALVE
RELIEF VALVE
B
N
G
F
E
D
C
M
H
C
B
M
OIL INLET
OIL OUTLET
STEAM OUTLET
Small hole on highest part of
pipe to prevent siphoning

Fig. 276F.

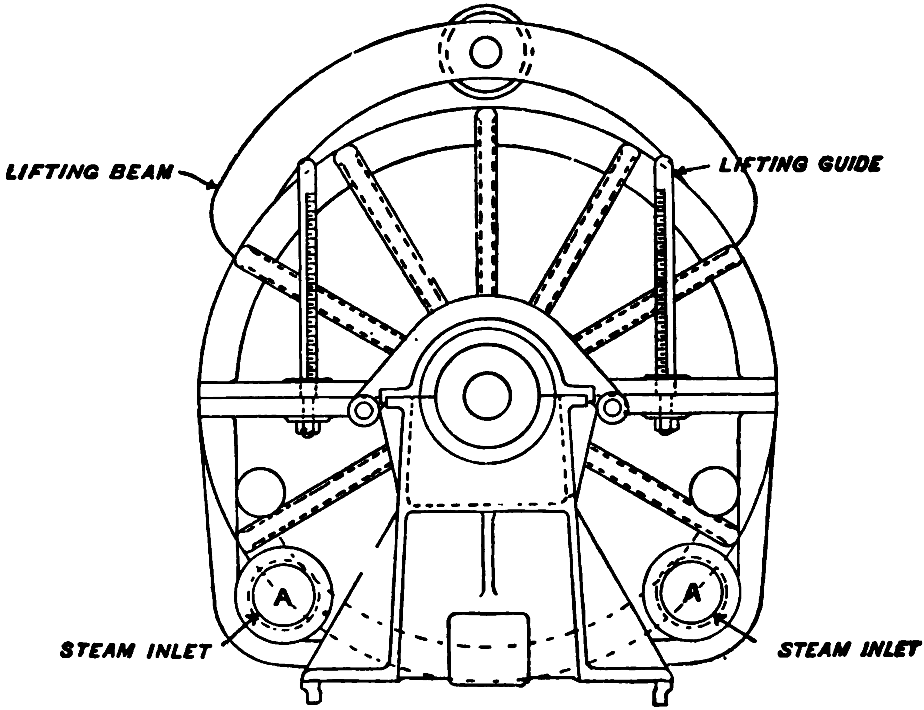


FIG. 276G.

three or four propeller shafts. In the latter case there are two high pressure ahead turbines, on two separate shafts, and the exhaust orifices H lead the exhaust steam from the high pressures to low pressure ahead turbines fitted on separate propeller shafts, one on each side of the ship.

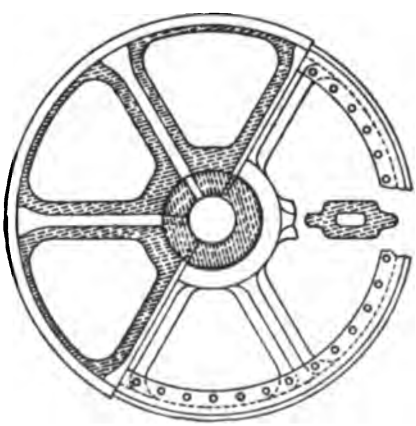


FIG. 276H.

Low pressure ahead and astern turbines.—The construction of the L.P. ahead turbine is shown in Fig. 276I and in the right-hand portion of Fig. 276J. In this turbine the steam received from the high pressure exhaust pipe enters through the inlet orifice which is shown in Fig. 276I, and further considerable enlargements of cylinder and expansion of steam take place, till exhaust to the condenser occurs at a very low pressure. The separate low-pressure turbine required for going astern is usually combined in the same casing as the low-pressure ahead turbine and

exhausts into the same orifice. This astern turbine is shown at the left-hand side of Fig. 276J. In this case the steam enters at the other end of the casing, and when going astern the direction of flow of steam along the turbine is opposite to that when going ahead. An elevation of the low-pressure ahead blades is given in Fig. 276L, which shows their number and general proportions at the different stages. All turbine casings are fitted with spring loaded relief valves.

Thrust forces.—The diameter of the parts N (Figs. 276F and 276J) called the dummies is proportioned so that the steam pressure on the

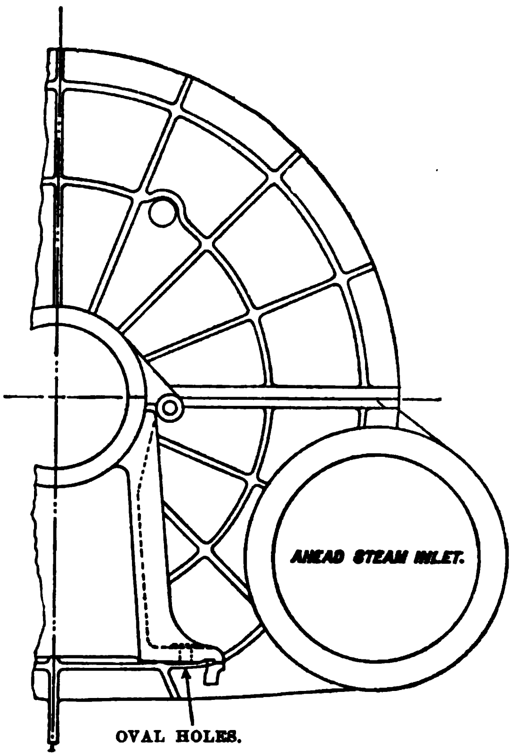


FIG. 276I.

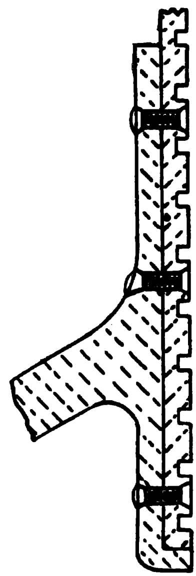
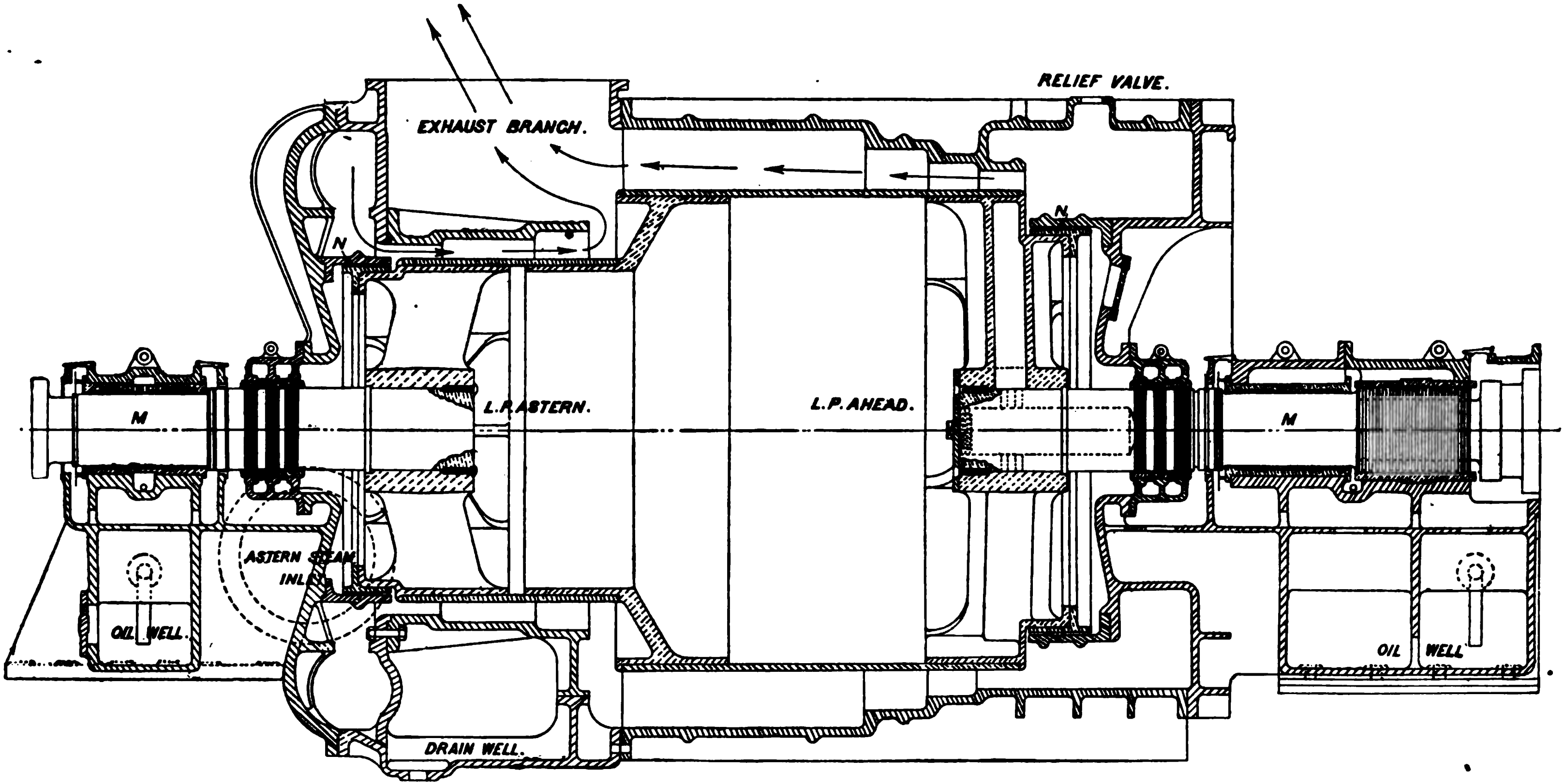


FIG. 276K.

parts of the rotors exposed to steam pressure, the pressure of the steam in the moving blades and the thrust transmitted from the propeller to the turbine shaft, will so far as possible mutually balance.

The forces in a fore and aft direction on shafts and casings due to the effect of steam pre sure and propeller thrust for ahead working may be conveniently summarised as follows, having in view the differences of steam pressure throughout the turbine casing, viz., a fairly high pressure at forward end of casing, a comparatively low pressure at after end and a pressure decreasing in gradual stages in passing through the blades.



RELIEF VALVE.
EXHAUST BRANCH.
N
N
L.P. ASTERN.
L.P. AHEAD.
M
M
ASTERN STEAM INLET.
OIL WELL.
OIL WELL.
DRAIN WELL.

FIG. 276J.

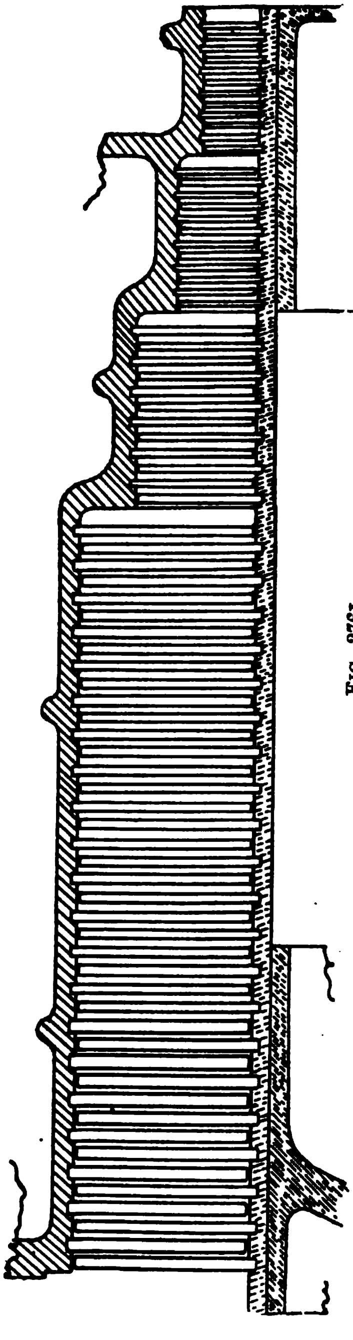


FIG. 276L.

Forces on Shaft :—

a. Thrust on shaft due to differences of steam pressure at successive rows of rotor blades in the aft direction.

b. Thrust due to differences of steam pressure on exposed parts of forward and after ends of rotor and dummy rings. The net effect is generally a steam thrust in an aft direction.

c. Propeller thrust in forward direction.

Forces on Casing :—

d. Thrust due to difference of steam pressure on exposed part of forward and after covers of turbine casings in a forward direction.

e. Thrust on casing due to differences of steam pressure at successive rows of casing blades in an aft direction.

It is evident that an exact balance of the forces *a*, *b*, and *c* on the shaft cannot be obtained at all powers or under such conditions as starting, head seas or winds, dirty bottom, &c., when the propeller thrust for the power developed is greater than usual, and there will therefore be at times a resultant thrust effect on the shaft, which may either be in the forward or after direction according to circumstances. This thrust is taken by a small external thrust block (Figs. 276M and N), so arranged that the bottom half takes any resultant forward thrust, while any resultant astern thrust is transmitted by the upper part or cap to the main casing by means of small screws, the position of which is indicated in Fig. 276N, and shown later, on the plan at B B in Fig. 277O, and these screws also provide for adjustment.

Supposing the thrusts under *a*, *b*, and *c* are mutually balanced, or in other words that the turbine shaft and rotor is balanced in a fore and aft direction,

then the actual thrust on the ship arises from the resultant of thrusts *d* and *e* on the casing, which is transmitted through the casing to the holding-down bolts and thence to the ship's structure. This thrust must, however, be equal to the propeller thrust since this is the only external force.

If the thrusts *a*, *b*, and *c* do not mutually balance, there remains an unbalanced thrust on the shaft, in a forward or after direction as the case may be, which is taken by the thrust block, and the thrust seating being secured to the turbine casing, the unbalanced thrust is transmitted to the ship structure through the casing holding-down bolts together with the resultant of steam thrusts *d* and *e* on the casing. The net thrust transmitted to the ship is as before, i.e. the thrust transmitted by the propeller. The holding down bolts of the turbine casing must therefore be well fitted to the seatings and of ample strength.

Ahead and astern dummy pistons.—As both the rotor and the dummy piston are hollow, any steam which may flow past the dummy piston will escape to the exhaust side of the turbine and constitute a source of waste. Special means have therefore to be provided to check the flow of steam past the dummy piston. Where an astern rotor is coupled to the same shaft and included in the same casing as the ahead rotor a corresponding astern dummy piston is fitted (see N at left of Fig. 276J), the steam-tightness of which has also to be assured. From considerations of longitudinal expansion, the methods of doing this differ in the two cases. For the ahead dummy, termed a contact dummy, rings are turned on it and corresponding collars fitted, the latter being made up of a succession of brass strips about six inches long let into grooves in the dummy piston case, caulked in, and finally turned to the section shown at the lower part of Fig. 276O; the final adjustment of the collars and rings is such as to leave a clearance of about $\frac{15}{1000}$ inch to $\frac{20}{1000}$ inch where indicated, the consequent repeated wire-drawing of the steam at these spaces and its subsequent expansion result in a fall of pressure as it travels outwards by which the extent of the leakage is very much reduced.

In order to reduce leakage to a minimum it is desirable and usual to fit this type of ring to the dummies of all ahead turbines; and in cases where two ahead turbines are secured to the same shaft, it is not possible to have a rigid coupling between the rotor shafts, due to the fact that the different rates of expansion under various working conditions would modify the dummy longitudinal clearance and possibly permit the rings to foul. Under these circumstances an expansion coupling of the type shown in Fig. 276Q is fitted between the rotor shafts of the two ahead turbines, and is such that, although capable of transmitting a turning moment from one shaft to the other, the two shafts can move longitudinally, within certain limits, independently of one another. It will be seen that when an expansion coupling is fitted a thrust block must be provided for each ahead turbine on the same shaft.

The astern dummy piston and casing are both fitted with short circumferential strips caulked in as before explained, thus forming alternate rings and collars; these are then turned to the shape shown in Fig. 276P, termed a 'radial dummy,' where it will be observed that the clearances at tips are small, involving the same wire-drawing and ex-

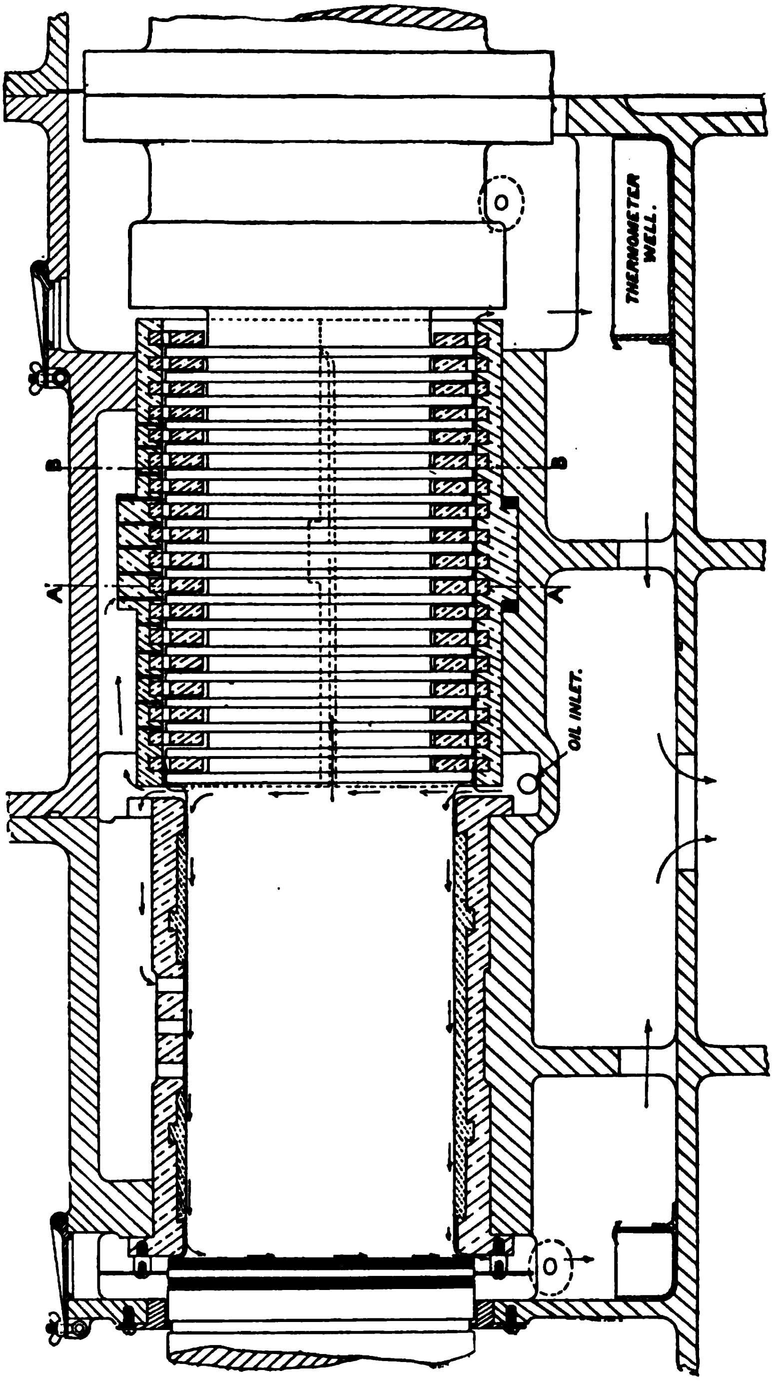


Fig. 27Cm.

pansion of the steam and continuous reduction of pressure as in the ahead dummy piston packing. Their arrangement admits of considerable latitude in the matter of relative longitudinal expansion between casing and rotor, which the ahead type does not.

Rotor shaft glands.—Steam-tightness has also to be secured at the glands of the rotor shaft. These glands may be subject to steam pressure, when the tendency is for the steam to pass them and flow out into

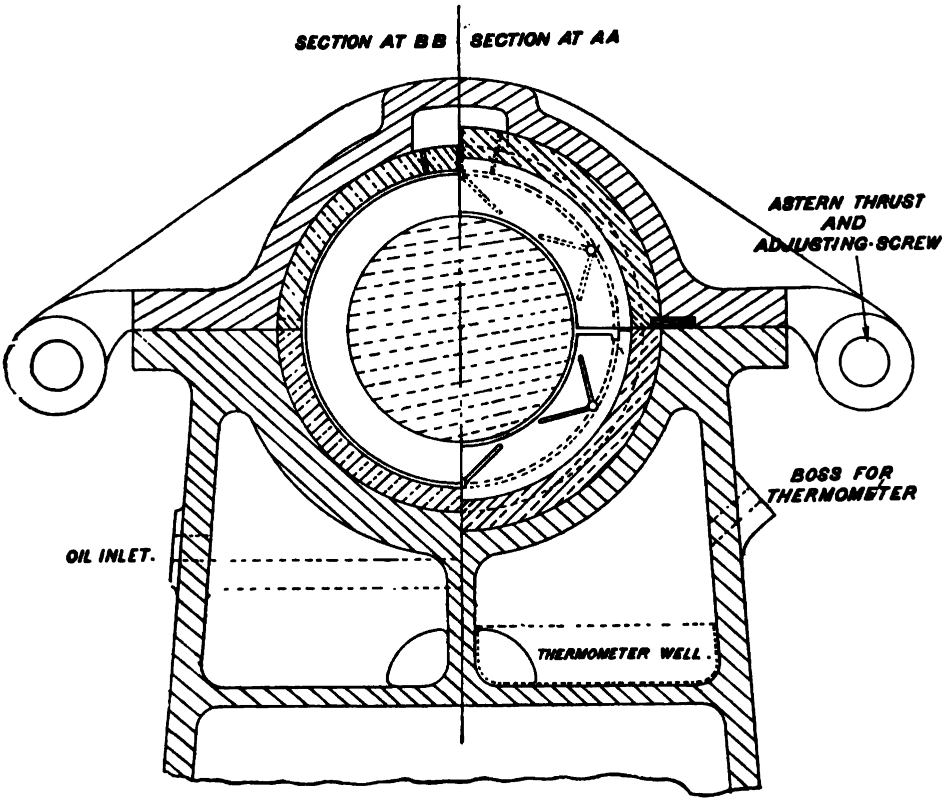


FIG. 276N.

the engine room, or they may be subject to a vacuum, when the tendency will be for air to enter the turbine and spoil the vacuum, or the same gland may, under the varying conditions of working, be subject either to steam pressure or a vacuum. The method originally employed was to turn a number of rectangular grooves on the spindle and fit these with Ramsbottom rings, the whole rotating in an accurately bored gland. A break existed in the continuity of these grooves, the gap formed being put into communication, by a number of holes through the gland, with an annular space in the body of the casting.

If the gland has to deal with steam pressure inside the turbine,

this annular space is put in communication, by means of a pipe and valve, with some part of the turbine system where a suitable lower pressure exists; such steam as may escape past the first series of rings is then 'leaked off' until the pressure is sufficiently reduced, gauges being fitted to regulate the pressure to about that of the atmosphere,

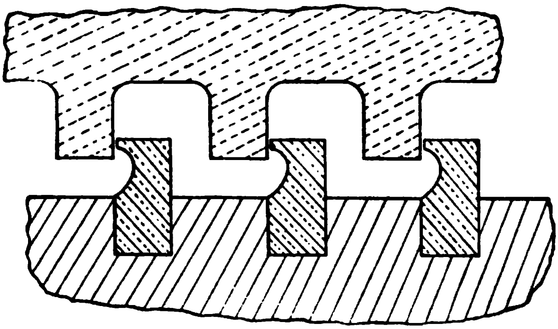


FIG. 276O.

as a slight weep of steam into the atmosphere becomes observable. If a vacuum exists inside the gland, the annular space communicates by a pipe and valve with a source of pressure, and sufficient steam is admitted to produce atmospheric pressure as in the former case, the existence of this pressure preventing the ingress of air. If the gland is variously subject to both steam and vacuum a double valve box

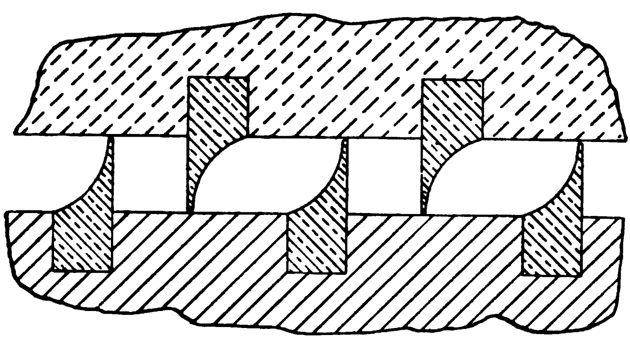


FIG. 276P.

is fitted, by means of which steam may be either 'leaked off' or a pressure introduced as requisite.

When the difference of pressure between the inside of the turbine and the atmosphere is great, the side pressure on these rings may be excessive, in which case the rings were subdivided into three or four groups, and the pressure between the groups suitably graded either by admitting or leaking off steam to the spaces between them. Pressure

gauges are connected to the annular recesses around the gland, by means of which a suitable adjustment of pressure is arrived at.

Experience has shown in some cases that with the pressure and sizes employed the Ramsbottom rings became badly scored and torn, and to obviate this a combination of the radial dummy packing, with the Ramsbottom rings, has now been adopted. Provision is made so that the Ramsbottom rings can be lubricated both when the turbines are at work and when stopped.

A typical gland of this character is shown in Fig. 277 with three groups of rings, the inner two being of the radial, and the outer of the Ramsbottom type. Annular recesses in the casting communicate with the spaces between these groups of rings, and pipes and valves are fitted as already described for leaking off or admitting steam, pressure gauges being fitted for observing and adjusting the pressures as requisite. To prevent any steam leakage from escaping to the engine room, a hood B (Fig. 277), with pipe leading to an up-cast ventilating hatch, is fitted.

Bearings.—The bearings M take the weight of the rotor, and must be very carefully lubricated, owing to the high speed of rotation of the shaft. They are therefore supplied with oil under about 10 lbs. per square inch pressure by a pump, the oil inlet and outlet orifices being indicated in Figs. 276M and N. The oil passes through a cooler surrounded by water, which is a small surface condenser, on its way to the bearings, or cooling coils are fitted in the oil-receiving tanks. Each bearing is fitted with a pressure gauge on its oil-supply inlet, a thermometer well and thermometer, a test-cock on the inlet passage discharging by means of a broken pipe to the outlet pipe, also sight holes and corresponding doors, by means of which the passage of oil can be readily tested. The bearings are lined with white metal, but in order to prevent the rotor dropping and injuring the blades in case of a defect in the white metal, each bearing is fitted at each end with a broad strip of gun metal which is just clear of the shaft, as in Fig. 276M, or with some corresponding device. An oil ring or baffle is fitted at each end between the oil chamber and the turbine to prevent any oil creeping along the shaft to the turbine or any escape of steam from the gland mixing with the oil. The direction of the flow of oil from the oil inlet orifice through the main bearing on one side and thrust bearing on the other to the outlet at the bottom of the casting is indicated by small arrows on Fig. 276M.

As it is most important that the wear down of these bearings should be readily ascertainable, the following gauges are supplied :—

(*a*) Bridge gauge, Fig. 277A, which is for use when the engines are stopped. It can be secured to the framing, and feelers are inserted between the face of the hardened steel pin A and the shaft. Comparison of this with the original measurements indicates the wear.

(*b*) Gauges for ready application whilst shaft is in motion. The method of testing can be readily understood from reference to Fig. 277B, which shows the sight-hole cover removed and the gauge in position for measurement.

Speed of rotation.—The speed of rotation of the turbine with its shaft and propellers is considerable (see Table), especially with the smaller diameters, as a high speed of rotation is required to obtain the speed of periphery necessary for efficiency ; 900 to 1,100 revolu-

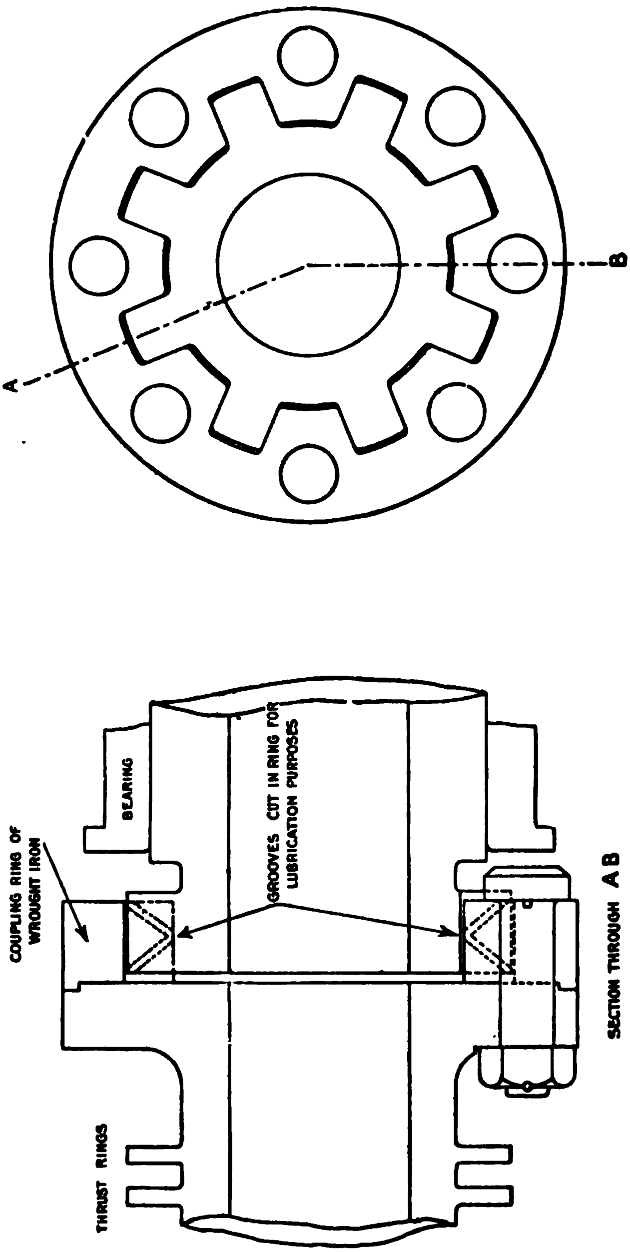

A
B
BEARING
COUPLING RING OF WROUGHT IRON
GROOVES CUT IN RING FOR LUBRICATION PURPOSES
THRUST RINGS
SECTION THROUGH A B

FIG. 276a.

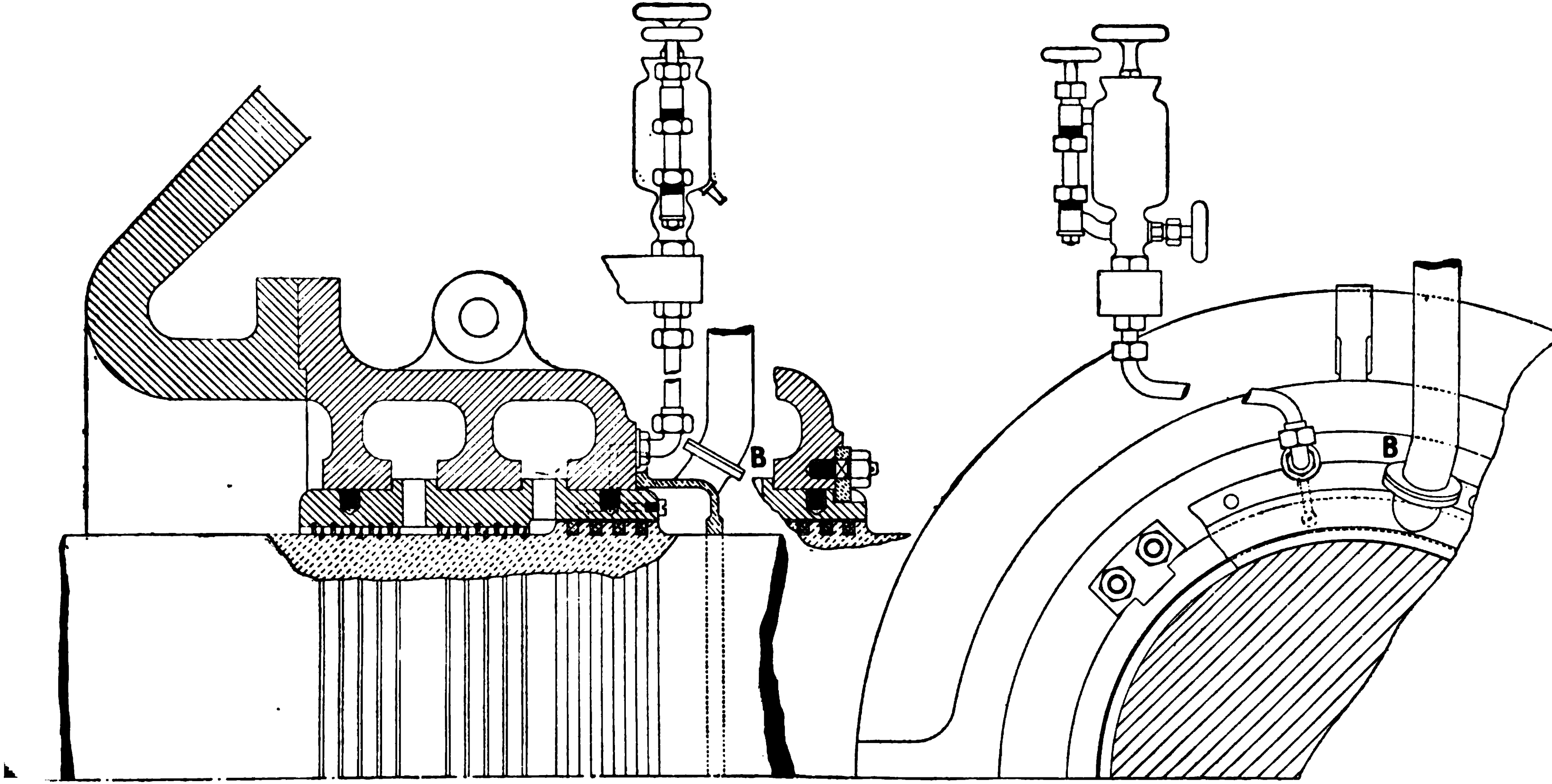


Fig. 277.

Ship	Steam pressure in lbs. per sq. in.	External diameter rotor barrel in inches		Mean diameter at blade centres in inches		Revolutions per minute at full power		Mean blade velocity in feet per second	
		H.P.	L.P.	H.P.	L.P.	H.P.	L.P.	H.P.	L.P.
Viper and Cobra . .	250	27	38	29·5	44·2	1,050	1,050	185	203
Velox	250	27	38	29·5	44·3	890	890	114	172
River Class destroyers .	250	28	40	29·8	43	940	940	122	176
Amethyst . . .	250	60	60	61·8	64·1	450	500	121	140
Dreadnought . .	185	68	92	70·9	99·2	320	320	99	138
Bellerophon Class .	170	68	92	72·0	100·2	320	320	100	139
Invincible Class . .	185	92	115	95·9	125·1	275	275	115	150

tions per minute is the speed adopted in the torpedo-boat destroyers built for the Royal Navy.

Balancing.—In order to reduce vibration to a minimum, and to avoid whirling of the turbine spindles, special attention has to be given

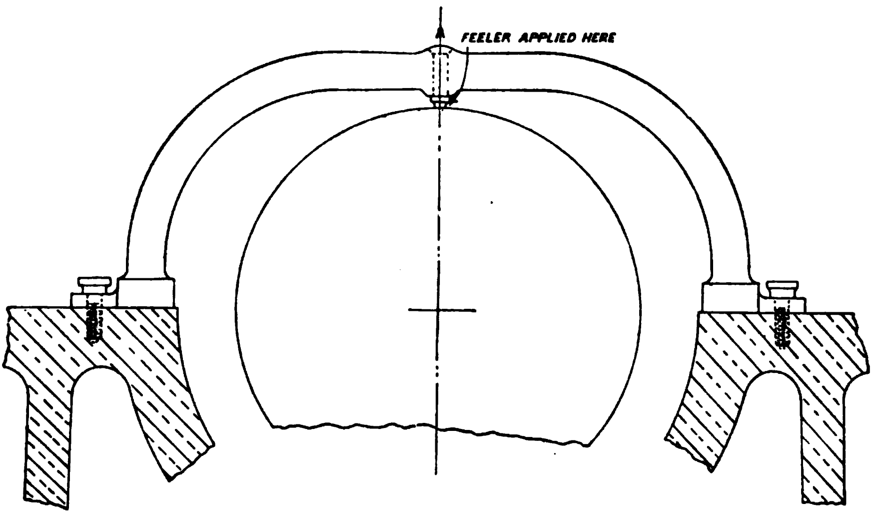


FIG 277A.

to the balancing of the revolving parts. The rotors are balanced statically on narrow truly surfaced edges (see Fig. 277c), weights being affixed to the rotor wheel arms or wheel casting rim as necessary for this purpose. The rotor is subsequently tested dynamically either by being rotated under steam in its casing, or by placing the rotor in an arrangement of bearings which are kept in place by springs, and rotating the rotor by means of an electric motor when any further adjustments which may be required are made.

The rotors are run up to a speed 25 per cent. in excess of the maximum designed working revolutions per minute, and any correction necessary to balance it under these conditions is made.

Amount of steam expansion.—With this form of engine it will be seen that very considerable expansion of the steam is possible ; about 100 expansions can easily be arranged for, while there is no alternate heating and cooling of the surfaces exposed to the action of the steam,

so that an economical performance can be obtained. The amount of expansion given in any one turbine is, however, constant, as is also the annulus for the passage of steam—i.e. the space between the rotor

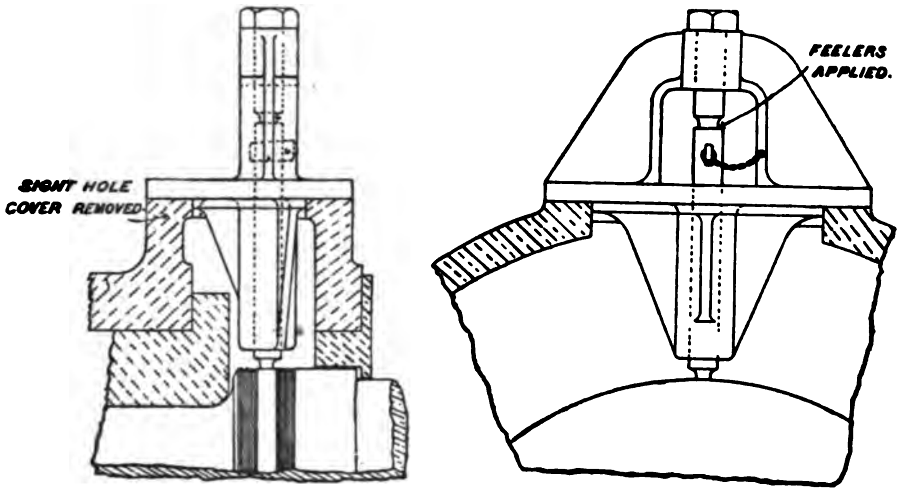


Fig. 277b.

and the casing; reduction of speed can therefore only be obtained by lowering the initial pressure of steam and consequently less benefit

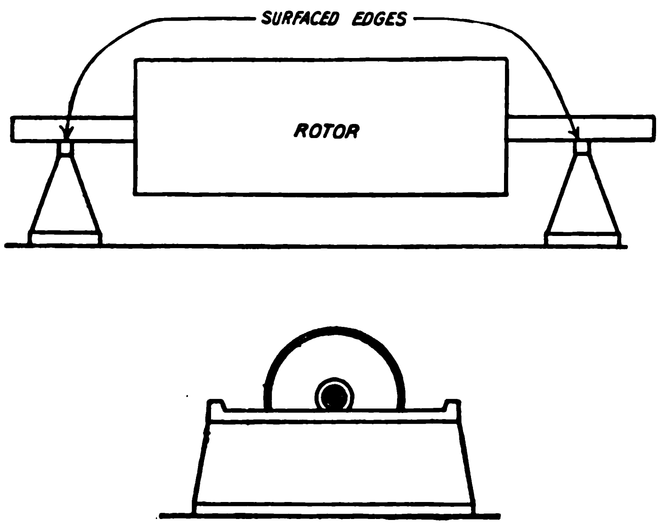


Fig. 277a.

from expansion is then realised, while as the revolutions decrease, the speed becomes less suitable for high efficiency, so that at low speeds the consumption of steam per horse-power may be expected to be more

in excess of the consumption at high powers than it is in the reciprocating engine, unless special arrangements are made. In the more recent ships less clearance than formerly between the tips of the blades and the drums and casings has been rendered possible by the tips of the blades being thinned as shown in Fig. 277J. This 'tipping' has the further advantage that should the points of the blades from any cause come in contact with the casing or drum, they will easily wear down and so prevent what might otherwise lead to a stripping of the blades.

Good vacuum.—A good vacuum is much more essential to the efficiency of the steam turbine than to that of the reciprocating engine. With the latter it is impossible to expand the steam right down to the pressure corresponding to the vacuum in the condenser, as the volume of L.P. cylinder thereby entailed would be prohibitive; it would also conduce to excessive initial condensation. This difficulty does not exist in the turbine, where expansion can be carried down to the lowest pressure obtainable. There is a great thermodynamic advantage in having a low condenser pressure which will be readily understood from a consideration of the following table:—

Vacuum in inches of mercury	Abs. press. in lbs. per sq. in.	Temp. of Boiling in °F	Proportionate increase of temperature in °F for an increase of 1 lb. pressure
—	214·7	387·5	
			·406
—	204·7	383·5	
			·420
—	194·7	379·3	
			·437
—	184·7	374·9	
			·458
—	174·7	370·3	
			·477
—	164·7	365·6	
20	4·90	161·5	
			10·4
24	2·94	141·0	
			14·2
25	2·45	134·0	
			17·1
26	1·96	125·6	
			21·0
27	1·47	115·3	
			28·3
28	·98	101·4	
			38·3
28½	·73	92·0	
			51·8
29	·49	79·3	
			82·4
29½	·24	59·1	

The following comparison also serves to illustrate the greater advantage of an improvement in vacuum as compared with an increase in boiler pressure. The liberation of heat energy when steam expands adiabatically from 170 lbs. gauge pressure down to a vacuum of 27 inches, is 278·1 B.T. units per lb. If the vacuum be increased to 28 inches,

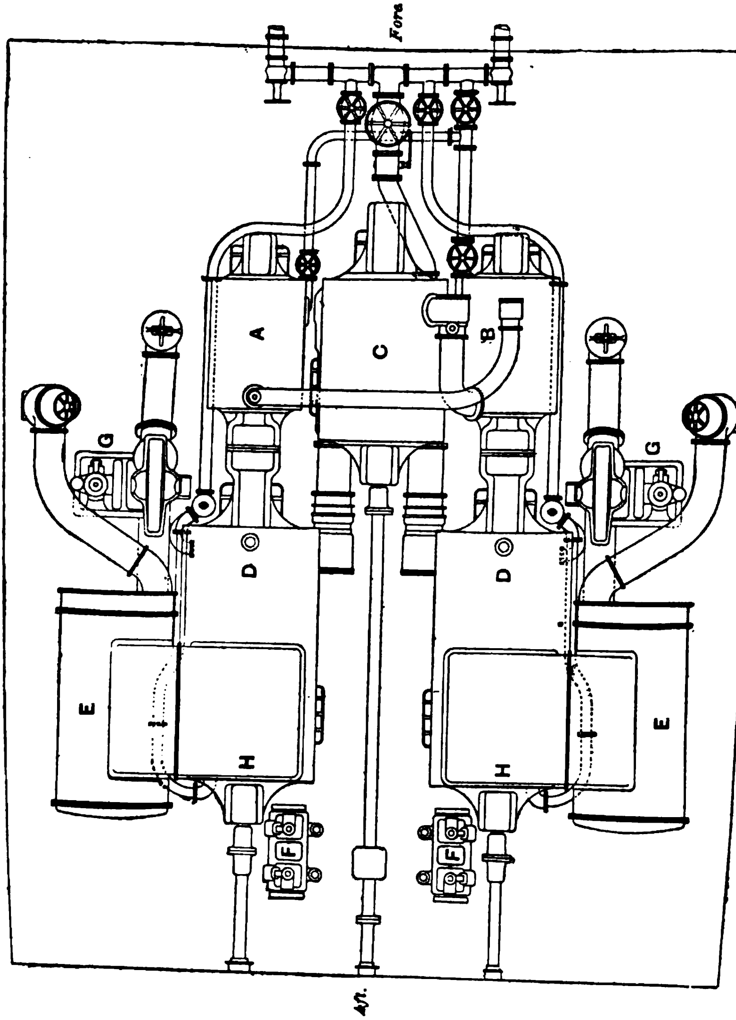


Fig. 277d.

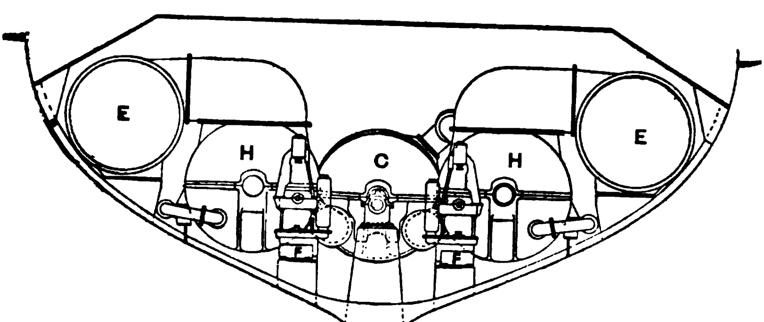


Fig. 277e.

the heat energy liberated is 293·7 B.T. units per lb., an increase of 5·6 per cent. If the pressure be increased to 190 lbs. by gauge, the vacuum remaining at 27 inches, the liberation of heat energy is 284·7 B.T. units per lb., or an increase of only 2·3 per cent. In view of the amount of expansion which is practicable in a turbine, the relative importance of increase of pressure in a turbine installation is much less than it would be in the case of a reciprocating engine.

The condensing plant therefore becomes an especially important factor with turbines, and usually leads to the provision of increased cooling surface per I.H.P., larger circulating and air pumps and special appliances for exhausting the air from the condensers, such as dry air pumps or Parsons' vacuum augmenters, described in Chapter XX.

Arrangement of turbine machinery on three shafts.—An arrangement of a set of turbine machinery in a ship is indicated by Figs. 277D and 277E, which show the engines of H.M.S. 'Amethyst.' There are three screw shafts in this case, on the middle one of which is the high-pressure turbine C to which boiler steam is admitted, and from which, after passing through the numerous rows of fixed and revolving blades, the steam is passed to two larger turbines, D, fitted one on each of the wing shafts, where it passes through another series of fixed and revolving turbine blades, and has its pressure reduced to a very small quantity. It is then exhausted from each of the low-pressure turbines into the condensers, E, which are provided with independent air-pumps, F, and the usual circulating pumps, G. When steam is admitted to the high-pressure turbine it proceeds to the two low-pressure ones, and therefore all three shafts revolve.

Manœuvring arrangements.—This being so, special arrangements are necessary to enable the port and starboard shafts to be stopped or worked ahead or astern independently when manœuvring or going in and out of harbours. For this purpose a separate manœuvring steam-pipe is led to each of the low-pressure turbines which enables them, when the steam to the high-pressure turbine is shut off, to be worked ahead independently of any other turbine. As the turbine can only be worked in one direction, as explained previously, a separate turbine with blades curved in the reverse direction is fitted on each wing shaft; these can be worked quite independently to give the ship stern way. Sometimes these reversing turbines are in a separate casing, but in the example illustrated the reversing turbine is fitted at H, at the after end of, but in the same casing as the low-pressure turbine, the steam for reversing entering at the after end through the pipe shown, and exhausting into the same exhaust pipe as for ahead working.

Cruising turbines.—For war ships in which most of the steaming is done at a comparatively low speed, it would not, as previously explained, be economical to admit steam direct to the large high pressure turbine at these speeds, so that additional smaller turbines called the 'high-pressure cruising' and sometimes an 'intermediate pressure cruising' are fitted one on each of the wing shafts but forward of the two low-pressure turbines. At the lowest powers boiler steam is only admitted to the high-pressure cruising turbine A; it proceeds thence to the intermediate cruising turbine B, and through the high-pressure turbine to the two low-pressures and condenser, the total expansion being very considerable. When higher powers are required than can

be obtained by this means, a steam-pipe supplying boiler-steam direct to the larger 'intermediate cruising' turbine can be used, while for greater powers steam is admitted direct to the 'high pressure' turbine.

To prevent the steam entering the 'H.P. cruising' turbine when steam is being admitted direct to the 'I.P. cruising' turbine, non-return valves are provided in the exhaust pipe from the 'H.P. cruising' turbine, and for similar reasons non-return valves are also provided in the other exhaust pipes between the various turbines. The final ones on the exhaust pipes between H.P. and L.P. turbines prevent steam entering the other turbines when working the low-pressures independently when manœuvring, or when using the reversing turbines.

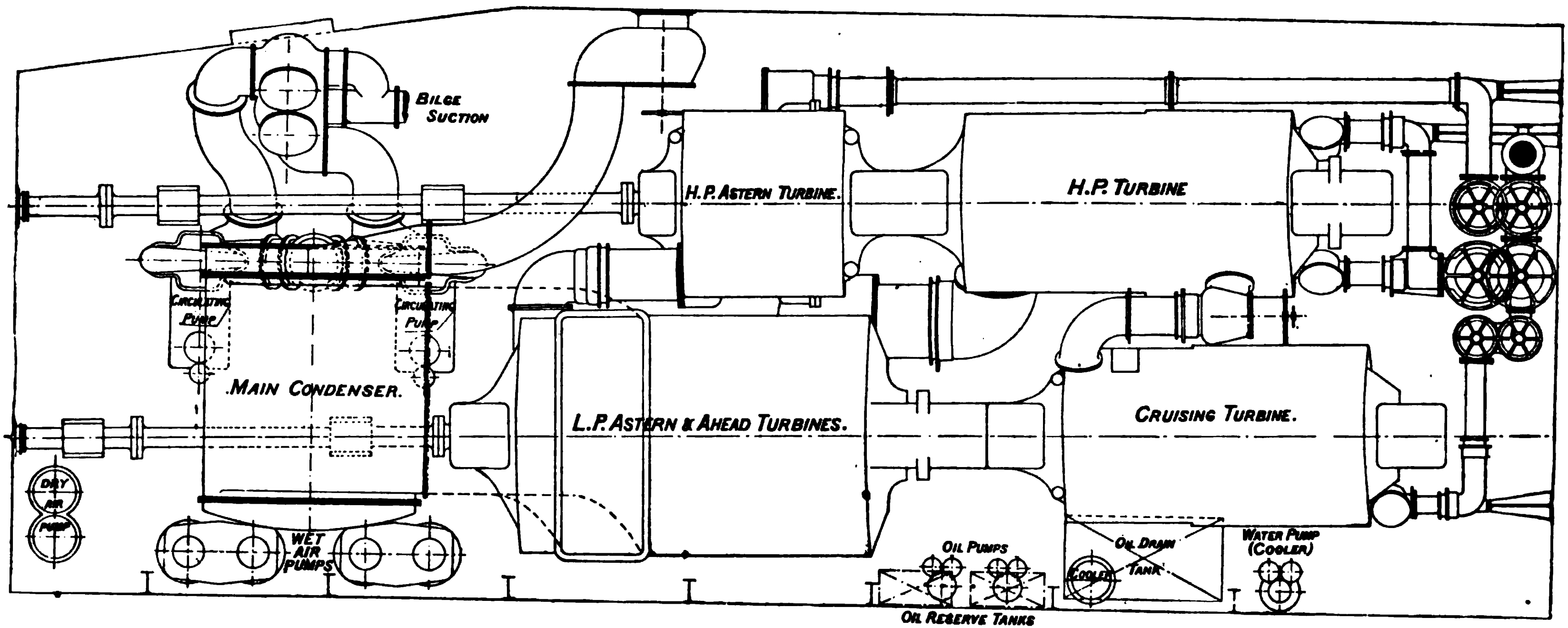
There is no doubt, however, that these cruising turbines add complications to the machinery, and as they are not always in use, they are somewhat more liable to accident than the main turbines, and for these reasons they have been omitted in recent British war-ships except where they are fitted in series, in which case the gain in economy is considered to be worth the extra complication.

Bye-pass arrangements.—To increase the power obtained from the cruising turbine a bye-pass valve is frequently fitted by which steam can be admitted at an intermediate stage of the expansion. This arrangement has also sometimes been fitted on the high-pressure ahead turbine in installations where the boiler-pressure was considered to be ample. Increase of power by this means is obtained at the expense of economy. This bye-pass arrangement is shown in Fig. 276F.

Description of four-shaft arrangement.—Fig. 277F shows an installation with four shafts divided among two independent engine rooms, and is that usually supplied to ships of great power. On each side of the ship there is a high-pressure turbine on one shaft exhausting into a low-pressure turbine on another shaft. For going astern a separate high-pressure astern turbine is fitted on one shaft, exhausting into the low-pressure astern turbine, which is combined in the same casing as the low-pressure ahead turbine. On this latter shaft is also a 'cruising' turbine for low power working, which, when being used, exhausts through the high-pressure ahead turbine to the L.P. ahead turbine and condenser.

It will be seen that there are three steam valves and pipes, each valve having a shut-off guard valve attached. One steam pipe leads to the high-pressure ahead turbine and branches into two smaller pipes leading to each side of the turbine. Another steam pipe leads to the cruising turbine, and an independent one leads to the high-pressure astern turbine. An expansion coupling of the type shown in Fig. 276Q is fitted to the shafts between the L.P. and cruising turbine for reasons previously given. The coupling between the H.P. ahead and H.P. astern is solid, as the dummy of the astern is of the type indicated in Fig. 276P. The remaining arrangements will be understood from the explanations given above and on the drawing.

Methods of construction.—The rotor drum is of forged steel generally rolled from the solid; this drum is turned internally and is shrunk on to the rims of the wheels, being also secured by screwed pins with their ends riveted over as indicated in Fig. 276K. The wheels are generally steel castings, formed as in Fig. 276H, though in some cases forgings are fitted. The wheels are shrunk on to the rotor shafting, additional security in each case being provided by screws screwed half in each as shown in Figs. 276F and 276J, or by similar fittings.



BILGE SUCTION
H.P. ASTERN TURBINE.
H.P. TURBINE
CIRCULATING PUMP
CIRCULATING PUMP
MAIN CONDENSER.
L.P. ASTERN & AHEAD TURBINES.
CRUISING TURBINE.
DRY AIR PUMP
WET AIR PUMPS
OIL PUMPS
OIL DRAIN TANK
COOLER
WATER PUMP (COOLER)
OIL RESERVE TANKS

FIG. 277F.

The drums are turned externally and grooved, the sides of the grooves being serrated with the object of affording a hold to the blades and packing pieces which fit these grooves (Fig. 277I).

The external casing is generally of cast iron, though cast steel has been employed ; this casing is in halves longitudinally with a stout flange on each half, by means of which the halves are bolted together.

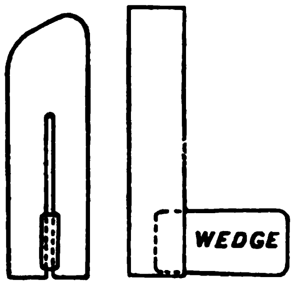


FIG. 277G.

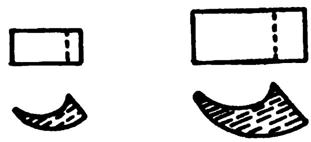


FIG. 277H.

The bolts require to be closely pitched, and in some cases a double row has been fitted. Where the dimensions of the casing are excessive, it is the practice to make it in pieces longitudinally, in which case circumferential joints occur as well as longitudinal. The flanges of this casing having been turned and planed as necessary for jointing the parts, the whole is set up in the boring machine and rough bored and grooved. It is then water-tested, and steam or otherwise annealed. Subsequently it is returned to the boring machine and the finishing cut taken. The object of this procedure is to correct any distortion that may arise from water-testing and the application of heat. This question of distortion of the casing is of considerable importance as affecting the economy of steam consumption, and likelihood of moving blades and casing coming into contact, whereby the blades might be stripped. For economy of steam it is very desirable that the radial clearances, i.e. the clearance between the tips of the blades and the adjacent parts, should be the smallest practicable, as the steam which flows past these openings performs no useful work, and for this reason 'tipping,' as referred to previously, has been introduced.

FIG. 277I.

Blading.—The operation of blading, i.e. securing the rotor and casing blades, in the Parsons' turbine is carried out as follows : in the rotor a stop of the form indicated in Fig. 277G is first secured in the groove by driving in the wedge shown, against this a packing piece similar to those shown in Fig. 277H is placed, and then a blade, this order being followed until there are a dozen or more blades and packing pieces in

position, then by means of a set having the end shaped to the curvature of the blades, these blades and packing pieces are closed up together by laying the set along the groove and driving it home by two or three blows with a hammer; additional blades and packing pieces are again loosely placed in position and the operation of closing up repeated, and so on until the groove is filled.

To adjust the blades to the correct distance apart at the tips and to ensure that they are radial and at right angles longitudinally, set

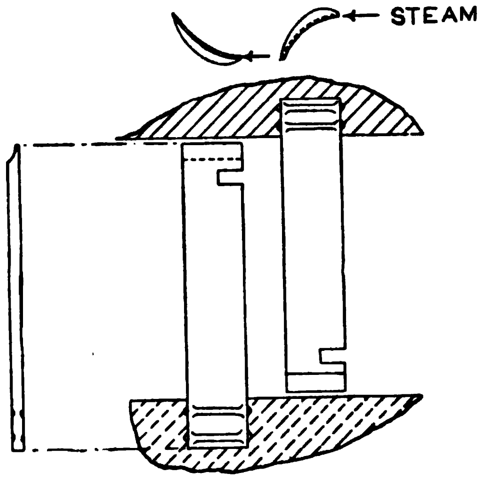


FIG. 277J.

squares are employed, those for the radial adjustment having the base cut to the circumference of the cylinder or casing from which the blades are set up. About every tenth blade is adjusted in this manner, those occupying the interval being trued up by a short steel straight-edge laid along the sides. The blades are secured in this position by a caulking tool, the caulking end being shaped to correspond with the

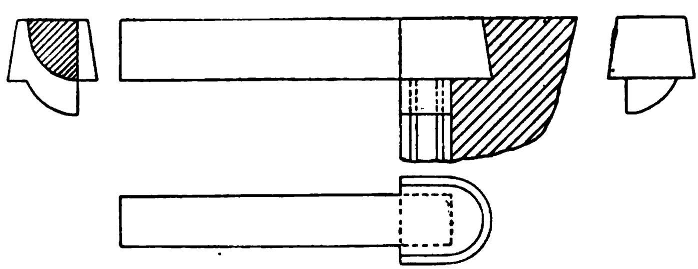


FIG. 277K.

packing piece. This caulking tool is inserted between the blades, and the packing piece expanded by giving it a few blows with a hammer; serrated sides formed in the groove in the rotor, Fig. 277I, and a couple of slight indentations stamped on the ends of the blades, Fig. 277J, serve

to hold the blades and packing pieces in position. To give still further security to the blades, they have a notch in their sides a short distance from the outer end into which a circumferential strip of brass is fitted, this strip is laced to the blades with fine copper wire, and finally, after being brushed over with a wash of borax solution, soldered with silver solder by a blow-pipe gas flame. Blades of considerable length have a second strip fitted, at about the middle of their length, those of 2 ins. and under have the fine copper wire omitted, whilst with blades under $\frac{7}{8}$ in. in length this form of security is entirely dispensed with.

In blading the casing, special provision has to be made at the longitudinal flanges to keep the blades in place, and for this purpose the ends of the grooves in the casing are recessed back into the flange by a milling cutter to the shape shown ; stops, one form of which is shown in Fig. 277K, projecting the length of the blades, have their ends fitted into these recesses, and are secured by swelling the ends out by hammering, the whole being finally filed flush with the flange. The

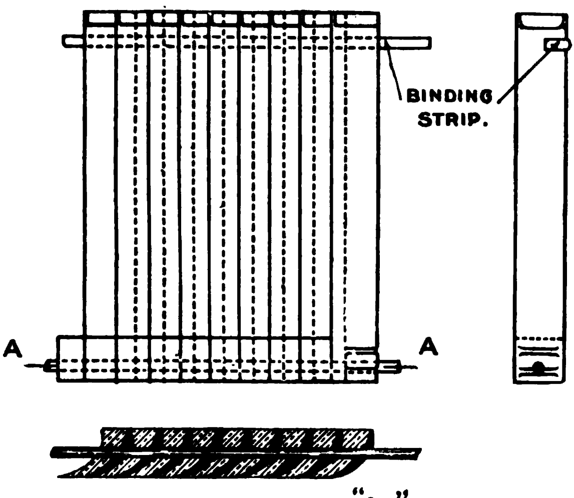


FIG. 277L.

curvature of the stops is different at the two sides of the casing, the necessity for which will be seen if the curvature of the blades is considered.

To facilitate the work of blading, both in new work and in case of repairs, various types of 'assembled' blading have been adopted. The most common is to set up a number of blades with packing pieces complete in a 'former,' these blades and packing pieces having a continuous wire passing through them, as indicated in sketch. These sections are caulked, as described above, in the 'former,' after which they can be transferred bodily to the turbine rotor or casing, when they are again caulked into the grooving. This form, Fig. 277L, is particularly convenient for reblading the upper halves of casings in a ship.

A fair amount of axial clearance exists between adjacent fixed and moving blades, varying from $\frac{3}{16}$ in. to $\frac{3}{4}$ in., nevertheless considerable care is required in lifting the rotor in and out, and in removing and replacing the top half of the casing; to ensure the blades against fouling, columns are fitted at the four corners of the flange of the bottom half of the casing, Fig. 276G, and corresponding guide-holes exist in the flange of the upper half. The columns are graduated, which enables the casing to be lifted equally at both ends by the beams shown, and the blades thereby retained parallel during the operation. Care must be taken in replacing the top half of the casing to see the joints of the dummy cylinder are correct, and methods of keeping these joints steam-tight are shown in Fig. 277M.

Expansion movements.—Reference has already been made to the annealing of the cylinder casing, which is carried out before the final cut is taken. This question of expansion and distortion due to the varying temperature at which the turbine works is of considerable moment. For example, taking the pressures in the main turbines of

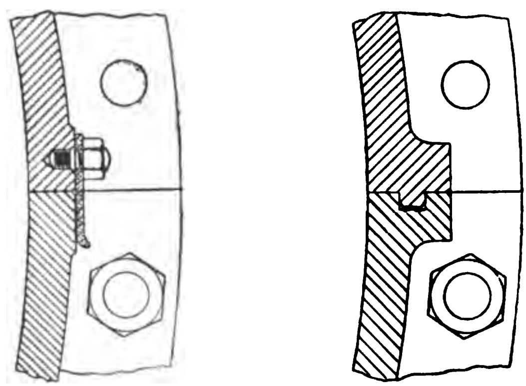


FIG. 277M.

the 'Amethyst' during her full power trial, we find that the H.P. turbine was subject to a temperature of 377° F. at the inlet, dropping to 270° F. at the exhaust, the corresponding temperatures for the L.P. turbines being 270° and 115° F. This change of temperature occurs over a length of from 6 to 8 feet, and remains practically constant; the radial expansion is consequently more at one end than the other, while at any point along the turbine the tendency is to expand less at the flanges than at the top and bottom: for this reason ample clearance must be allowed. The longitudinal expansion is often great, and the provision for the consequent movements has to be carefully considered in each particular case.

Referring to the four-shaft arrangement, Fig. 277F, as the condenser is fixed, and the eduction bend between the L.P. turbine and condenser is a very stiff irregular structure of steel plates and angles, this bend has no flexibility to take up expansion movements. It is

therefore necessary to reduce these movements to a minimum, and to meet this condition the foot at the after end of the L.P. casing is fixed. There is consequently a considerable movement due to expansion at the foot at the forward end of the L.P., and still more at the forward end of the cruising turbine, as the casings of the L.P. and cruising turbines are practically continuous.

In the case of the wing shaft, however, a different consideration arises; the casings are fixed to the ship at the feet between the turbines, this position being such that the movements of the sleeves in the expansion glands of the exhaust pipes from the ahead and astern turbines are reduced to a minimum, while at the same time the dummy movements, both in the case of the ahead and astern cylinders, are limited. The astern dummy packing, as can be seen from Fig. 276P, is pitched sufficiently widely as not to be affected by the longitudinal expansion allowed in the rotor and casing. A well-defined circumferential groove is made on the shaft, so that by measuring its distance from a fixed point on the H.P. astern casing the relative position of the dummy rings of the shaft and casing can be readily determined.

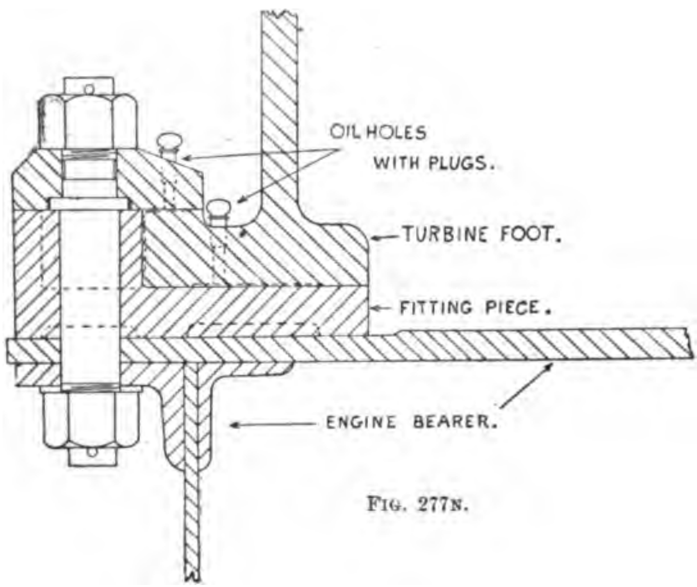


FIG. 277N.

Fig. 277N shows an arrangement for permitting the free movement of the supporting feet referred to above. This design allows free movement of the feet, whilst at the same time holding down the turbine to the engine seating. A similar provision is indicated in Figs. 276G and 276I.

Adjusting gear is required for moving the shaft in a fore and aft direction when performing the various operations required during the adjustment and fixing of the relative position of the rotor and the outer casing, which, owing to the small clearances being dealt with, is an

operation requiring great accuracy. Various arrangements have been fitted for this purpose, one of which, Fig. 277o, will be described. It consists of a semi-circular cap A grooved to fit the thrust collars and long enough to engage with five of the thrust collars when placed on the shaft. Stout lugs with screwed holes are arranged on each side of the cap, screwed bars are threaded through these lugs, and are carried at each end in bearings forming part of castings B, which can be rigidly secured to the lower half of the turbine casing. A worm and

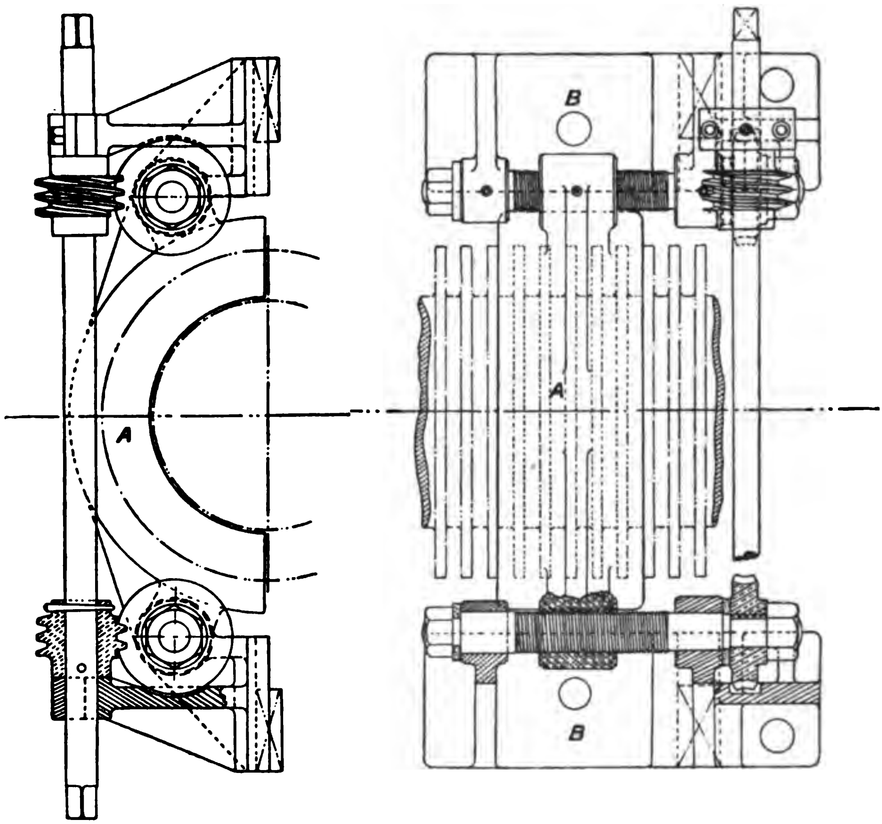


FIG. 277o.

gearing is arranged so that both screws are rotated together by the motion of the same handle, the screws being prevented from moving longitudinally by collars. When it is required to move the shaft longitudinally for adjustment purposes, the thrust block cover is removed and the gear fitted as shown, which enables the shaft to be moved to and fro in relation to the casing by the motion of the worm and handle.

Adjustment of 'contact dummy' clearances.—The 'contact dummy' clearance is indicated by a piece of metal fixed to the casing and inserted into a groove on the external part of the rotor shaft. This piece of metal is shown at C, Fig. 277Q, and is known as the 'finger

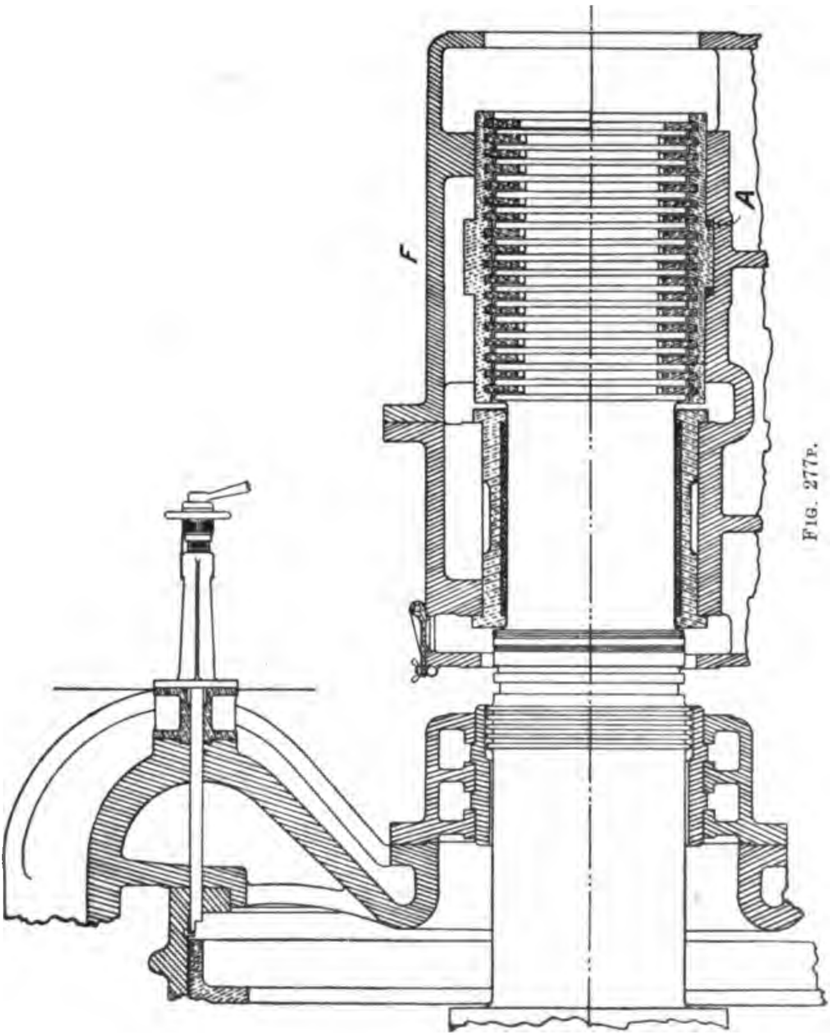


FIG. 277P.

piece.' The distance between the after face of the finger piece and the after face of the groove minus the gauging obtained when the dummy clearance is nil indicates the dummy clearance; i.e. the fore and aft distance between the rotating and stationary parts of the dummy.

The top and bottom brass rings of the thrust block are in contact

respectively with the after and forward faces of the rotor shaft thrust collars, excepting for the necessary clearance to ensure the free access of the lubricant, thus preventing any appreciable movement of the rotor longitudinally ; the bottom rings prevent the rotor coming for-

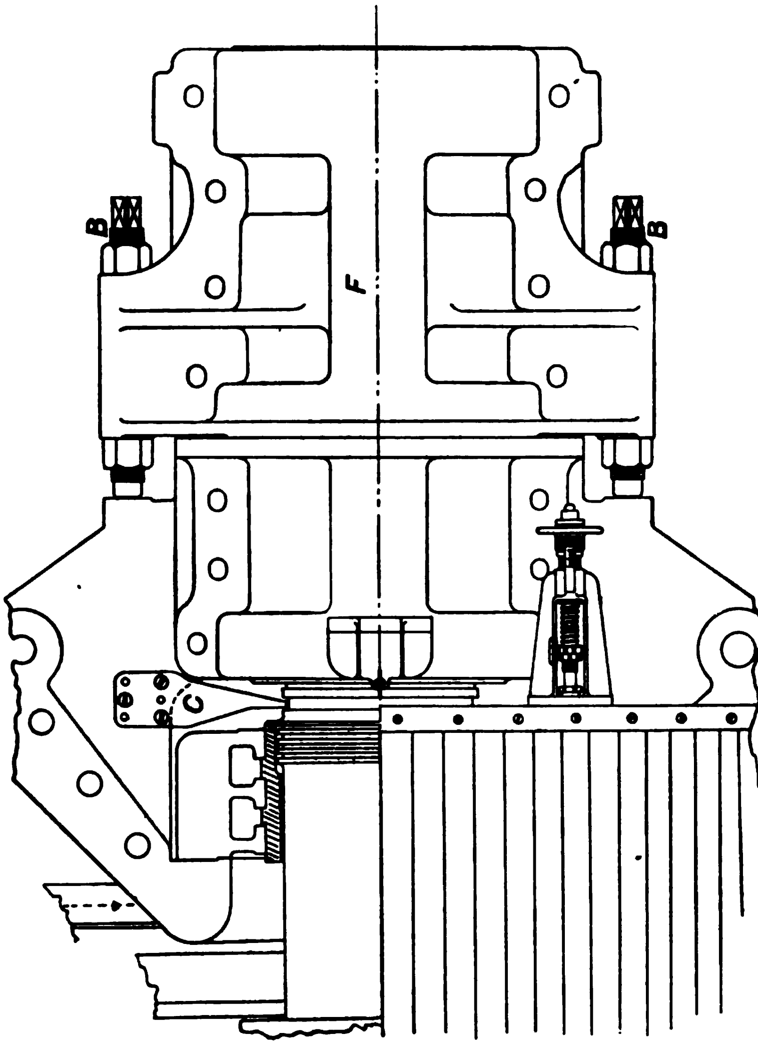
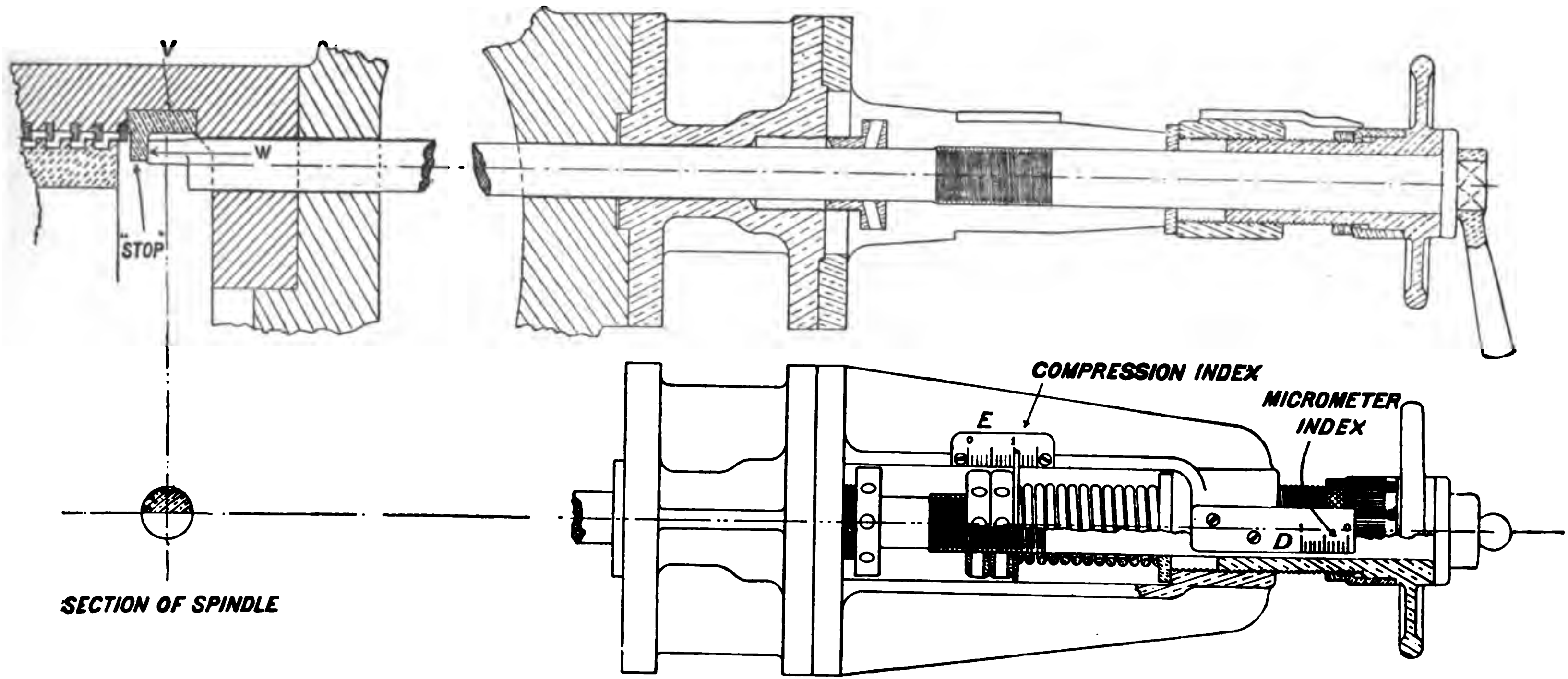


FIG. 277Q.

ward and fouling the dummies, and the top rings ensure that it does not move aft. The position of the bottom half of the block is determined by a semi-circular liner of square section shown at A, Fig. 277P, and that of the top half by means of adjusting screws B, Fig. 277Q.

The setting up and adjusting is performed as follows :—(1) Re-



V
W
STOP
SECTION OF SPINDLE
COMPRESSION INDEX
E
MICROMETER INDEX
D

FIG. 277B.

move the cap F with the brass of the thrust block; (2) remove the liner A; (3) draw the rotor forward by means of the gear provided, Fig. 277O, until the ahead dummies foul; (4) check the finger piece clearance and verify relatively to the original clearance at the finger piece when the dummy was touching; (5) ease the rotor back and insert a suitable liner A in the recess to give the necessary clearance; (6) bring the rotor forward again until the collars on shaft press on the thrust rings and gauge the clearance at the finger piece; (7) when the correct amount of clearance has been obtained, about $\frac{15}{1000}$ to $\frac{30}{1000}$, remove the adjusting gear, replace the cap with brass and tighten the nuts sufficiently to prevent the cap from lifting, but not from sliding (the holes in the cap are elongated); (8) turn the shafting and screw out the adjusting screws B equally until it is felt that the rotor shaft collars are pressing on opposite faces of the thrust rings; (9) tighten the nuts to retain the cap in this position and ease back the adjusting screws B to the amount allowed for lubricating purposes, about $\frac{4}{1000}$ to $\frac{6}{1000}$; (10) lock B in this position, ease off the cap nuts and move the cap F until the screws B are again touching the turbine end, the top collars have by this means clearance for lubrication purposes; (11) hand tighten all the cap nuts, and after a few revolutions under steam tighten them up hard.

These operations should be performed when the turbines are thoroughly hot.

Owing to the wide range of steam pressures, and consequent differences of temperature throughout the turbines, the actual dummy clearances are not always exactly in accordance with the gaugings at the finger pieces, the greatest differences occurring when the turbines are rapidly opened out to full power. To check these differences, arrangements (differing in detail) are fitted in some cases, at the steam end of cylinders, by which a spindle can be moved until its inner point comes in contact with the forward edge of drum. This spindle projects through the turbine casing parallel with the axis of the rotor, and gaugings are taken at its outer end. Care should be taken when this spindle is not in use that it is secured in such a position that its point cannot foul the drum. One of these gauges is shown in detail in Fig. 277R, in which case the pointer is put in contact with the fixed stop V and then turned half a revolution when gauging the position of the end of the dummy rotor, the readings being taken on the micrometer gauges D and E at the end of the spindle. This device eliminates errors due to possibility of wear on the end of the rod, as all measurements are virtually taken between the fixed face W and the end of the rotor drum.

The difference between the gaugings obtained (viz. when the turbine is hot and at rest, and when the turbine is rapidly opened out to full power) indicates the greatest error at the finger piece. When this amount has been ascertained, it should be allowed for at the finger piece when gauging the clearance, as at (4) above. The instrument should be arranged so that it is not possible to inadvertently screw it hard against the rotor drum. That in Fig. 277R is kept in contact by means of a spring.

Various arrangements of adjusting gear differing from that shown

on Fig. 277o have been fitted, but the same general principles are to be followed in making adjustments.

Auxiliary exhaust connection.—Provision is often made to allow the surplus exhaust steam from the auxiliary engines to be utilised in the turbines. A spring-loaded valve controlled by valves of the screw-down pattern is generally fitted on the auxiliary exhaust system having connections with the H.P. receiver, the turbines direct in a similar manner to the bye-pass arrangements shown in Fig. 276F, or to the L.P. receivers, as may be decided. The valve leading to the H.P. receiver is used in connection with the cruising turbine and that to the L.P. receiver when the H.P. ahead is in use without the cruising. Care must be taken that the exhaust spring-loaded valve on each condenser is screwed down to raise the pressure in the exhaust system so that it will exceed by 4 to 5 lbs. that which is required to pass into the turbines. Steam from the closed exhaust system when admitted to the turbines results in considerable economy, particularly at low powers. Care, however, must be taken that on receiving orders to stop, the closed exhaust system is immediately shut off from the turbines.

Curtis and Brown-Curtis turbines.—The Curtis turbine is of American origin, and was first fitted for marine purposes in that country. It is an 'impulse' machine, and in the early designs consisted of a series of wheels only, each containing a small number of rows of blades or buckets revolving between corresponding fixed blades on the casing. It has been developed in Great Britain by Messrs. J. Brown & Co.

Superior economical results are obtained by the combination of a number of the wheel stages with a drum stage at the low pressure end, and this latter type has been fitted in some ships.

Each wheel with its blades revolves in a separate chamber formed by fixed partitions or diaphragms, the steam expanding from one chamber to the next through inclined nozzles in the diaphragms, which direct the steam on to the revolving blades of the next wheel.

The fall of pressure of steam occurs in the nozzle, and is accompanied by a considerable increase in velocity of the jet which strikes the revolving blades and guide blades alternately, giving up energy to the former and driving the turbine wheels and shaft. The fluid pressure during the passage through the blades of each wheel remains constant until the succeeding nozzle is reached, when a further fall of pressure and increase of velocity occurs.

The drum stage consists of alternate rows of fixed blades in the cylinder casing and of moving blades on the drum as in the Parsons turbine, but the blading is of impulse type, i.e. each pair of fixed and moving blade rows may be regarded as a simple impulse stage, the fixed blades being the nozzles to the moving blades.

The angling of the blades is varied through each wheel and drum stage to allow for the falling velocity of steam through the stage, and Fig. 277s shows the angles of the nozzles, the fixed and revolving blades, the method of attachment of the nozzle frame to the diaphragm, the cylinder and wheel blading, and the attachment of the wheel rim to the disc and revolving blades.

No dummies are fitted in this type, and owing to the uniform

pressure throughout each stage the steam pressure on each wheel is balanced. There is also no unbalanced steam thrust on the wheel blades, and in order therefore to relieve the thrust block from the total propeller thrust the forward wheel of the drum stage is made solid, so that the rotor is subjected to a balancing astern thrust due to the steam pressure at the forward end of the drum stage.

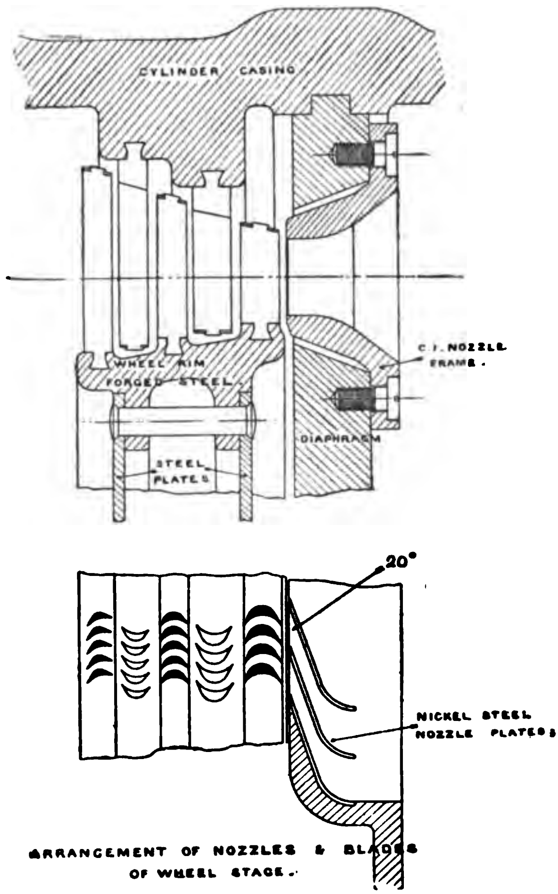


FIG. 277B.

As the cylinder end and spindle gland is subjected to the steam pressure in the first stage, i.e. the exit pressure of the initial nozzles, in order that this may be moderate, the pressure drop in the first nozzles is greater than in the remaining stages, and additional blade rows are necessary to utilise the higher steam velocity generated.

Owing to the higher pressures to which the spindle gland is subjected, compared with the Parsons turbine in which the gland

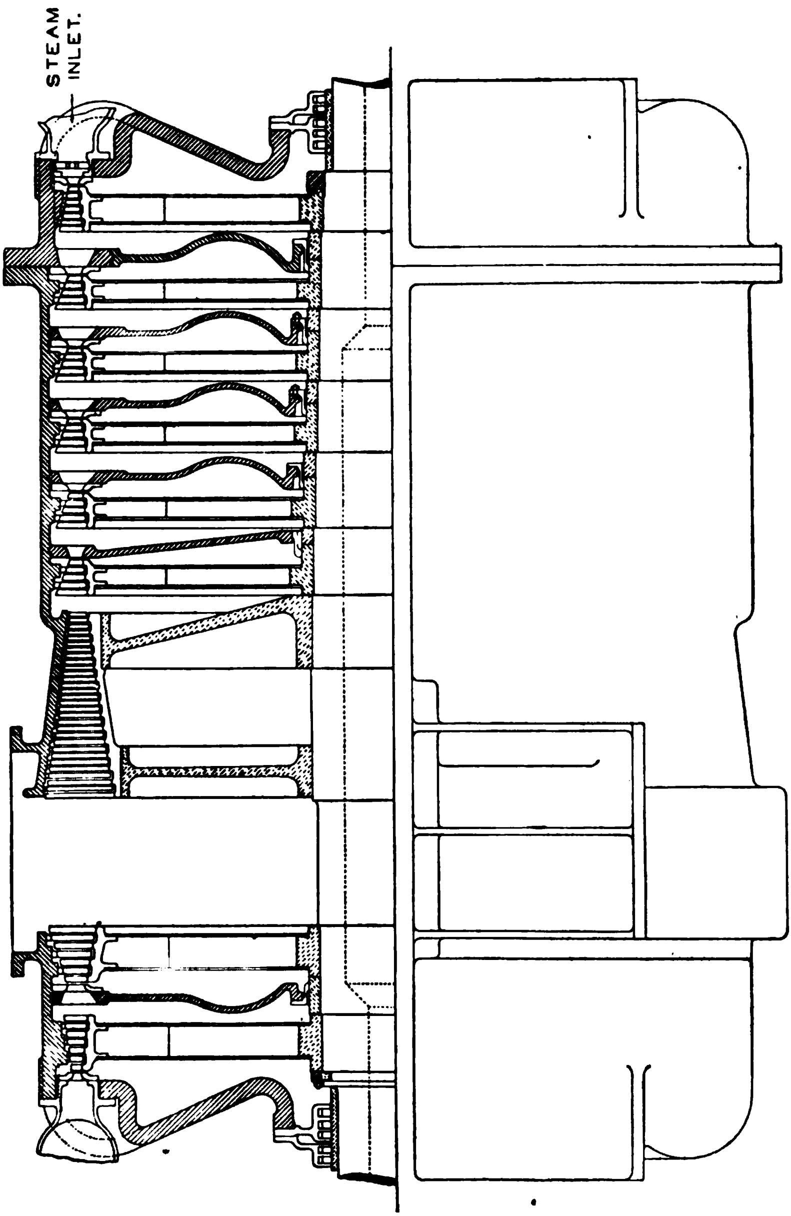


FIG. 277L.

is subjected to the exhaust pressure of the turbine only, a carbon block type of packing is fitted.

As indicated in Fig. 277U, there are about five rings of carbon segments in the cast-iron gland piece with an intermediate steam pocket, the carbon segments being held up against the spindle by circumferential springs. Leakage of steam between the wheel stages around the spindle at the diaphragm centres is prevented by serrated gunmetal bearing pieces carried in cast-iron gland pieces as indicated in Fig. 277V. It is usual to fit in addition one of the rings of carbon segments at the first diaphragm gland.

Nozzles.—The area through the nozzles of the fixed diaphragms increases with the volume of steam, the initial high pressure nozzles occupying only a small portion of the circumference, the nozzle arc

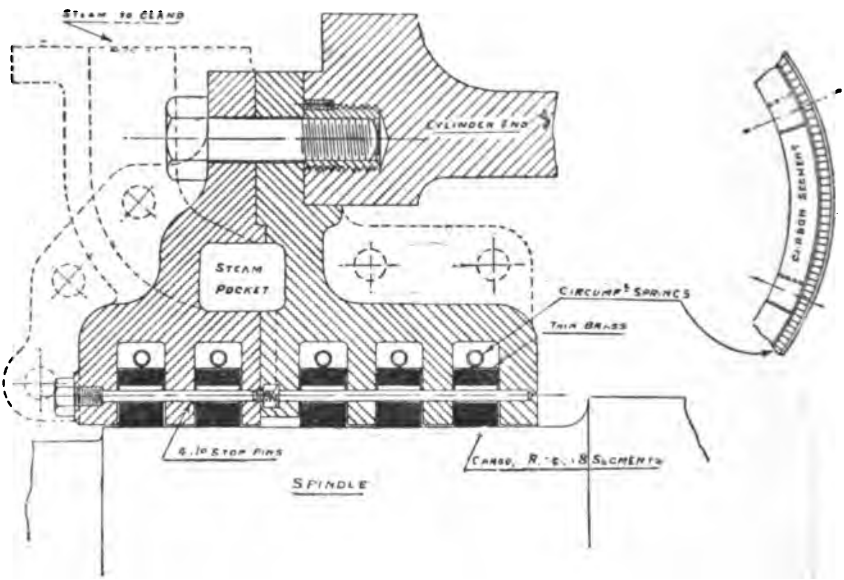


FIG. 277U.

gradually increasing until the whole circumference is used in the last wheel stages and the drum stage. As guide blades in the casing are only required in the vicinity of the nozzles, these also occupy a portion of the circumference corresponding to the nozzles. In Fig. 277T a section through the lower portion of the wheel stages would show no guide blades in the high pressure stages. One method of forming the nozzles is by nickel steel plating, securely held in cast-iron frames which are secured to the diaphragms.

The blades are of brass with dovetail roots fitting into undercut grooves in cylinder and rotor. The ends of the blades are riveted into shrouding bands both for cylinder and rotor blades, and owing to uniform pressure in each stage there is no need for small blade tip clearances in the wheel stages.

The diaphragms must be of stiff form, and excepting the first, which is solid, and is removed with casing end, they are in halves with spigoted horizontal joint, each half being spigoted at outer circumference into the cylinder casing.

The wheel centres are of forged steel, the spindle in boss being slightly taper, and the discs are of steel plate riveted to the boss and the steel rim carrying the blading. Steel distance pieces are fitted between the wheel bosses, the whole being secured by end nuts on the spindle. To facilitate the fitting of the wheels, and to give necessary stiffness towards the centre of the span between the bearings, the spindle is gradually stepped up in diameter from the ends. Owing to

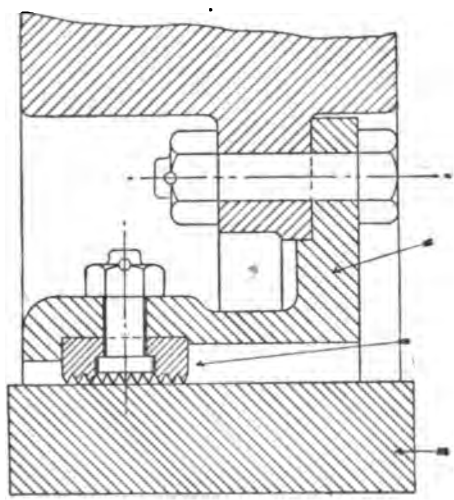


FIG. 277V.

the form of the cylinder casing a drain is necessary on each wheel stage.

Each turbine engine is complete on a separate shaft, the fitting of turbines in series on separate shafts being confined to the Parsons type, so that installations of Curtis turbines consist usually of turbines on two independent shafts, one on each side of the ship, the astern turbine being fitted in the same casing, as indicated in Fig. 277T. In recent designs a drum stage has also been added to the astern turbine.

The turbines run at lower revolutions than those of the Parsons type, and they are therefore larger, and the parts to be lifted and handled on board ship are bulkier and heavier, which is a disadvantage, but the two-shaft arrangement generally gives a more convenient engine-room, and the larger propellers running at lower revolutions give higher propulsive efficiency. Superheaters are often fitted to the boilers for these turbines. With moderate superheat the steam loses its superheat in expanding through the first nozzles, and the casing is subjected to steam at ordinary temperatures only.

Parsons combined impulse and re-action turbine.—An initial impulse stage has been fitted to the H.P. ahead or cruising turbine in some recent Parsons' turbine installations, the considerable pressure drop in this stage allowing of satisfactory economy with shorter turbines, and the whole of the expansion to be economically carried out on one shaft. The astern turbine on each shaft similarly has an impulse stage. Fig. 277w indicates the arrangement of the impulse stage.

To give greater economy at low powers the first reaction stage may be for cruising speeds only, the steam from the impulse stage being bye-passed to the second or full power reaction expansion.

Geared Turbines.—By the intervention of reducing gear between the turbine and the propeller shaft, high revolution and consequently

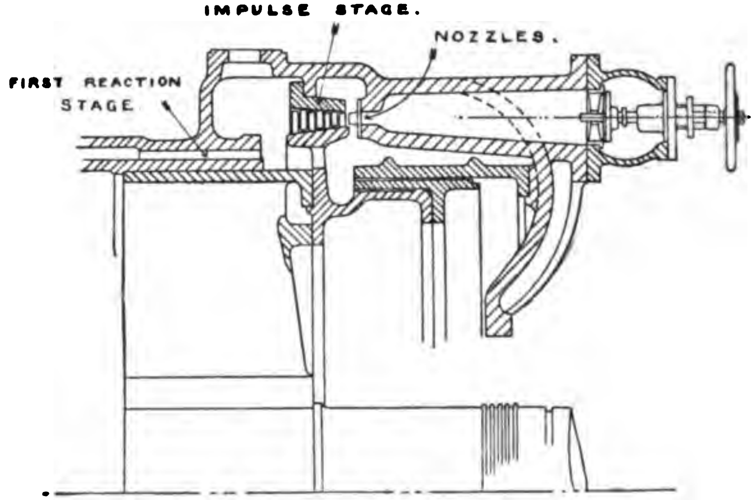


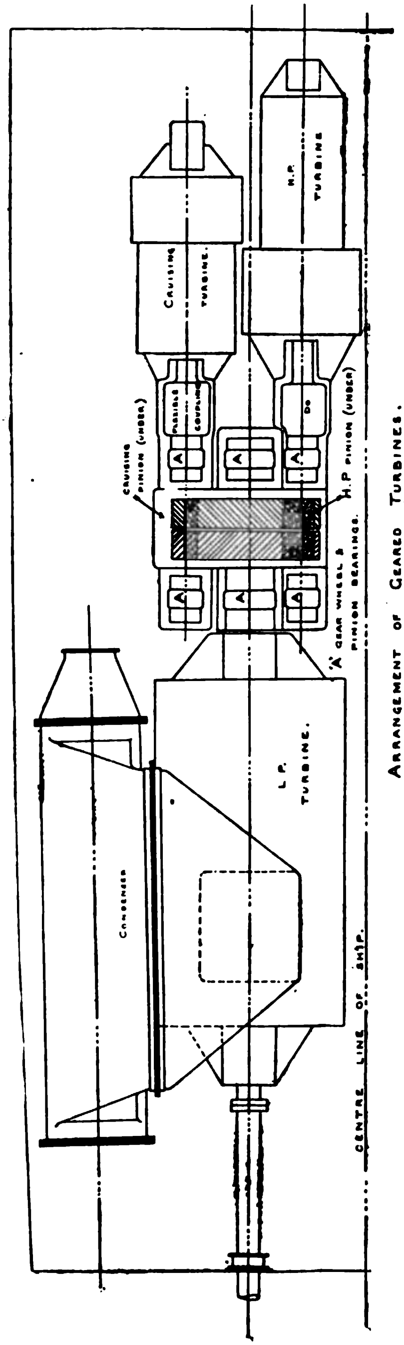
Fig. 277w.

increased economy with moderate size of turbines together with low revolutions and improved propeller efficiency are obtained.

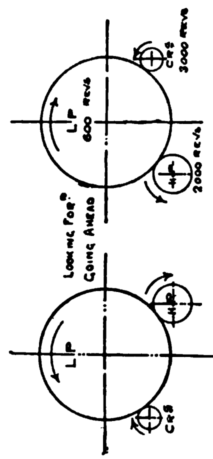
In view of the satisfactory result of trials in the experimental cargo boat 'Vespasian,' fitted with reduction gearing in ratio about 20 to 1, to give about 70 revolutions per minute of the propeller, this gear is being installed in some torpedo-boat destroyers. The general arrangement of the turbines and gear wheels is shown in Fig. 277x, each wheel and pinion having double helical gear in order to eliminate end thrust.

The large gear wheel on the L.P. turbine is built up of forged steel, and the pinions on the cruising and H.P. turbine shafts are of nickel steel. The gearing is enclosed in a cast-iron case, forced lubrication being fitted to all bearings, and an oil-spray arrangement for the gear wheels.

Other turbines.—Similar turbines to the preceding 'combined' types are also in use, the A.E.G. turbine being largely used with



ARRANGEMENT OF GEARED TURBINES.



GEARING DIAGRAM

FIG. 277E

success in Germany, while the Zoelly turbine is also being introduced. These differ only in detail from the types described in previous paragraphs.

Warming up.—Great care should be taken so as to uniformly warm up the turbine rotors and casings when preparing for steaming. Steam from the main regulating valves should be allowed to gradually pass through the turbines, and the positions of the steam and exhaust pipe connections being in general such that when steam is admitted through them with the turbines at rest it will not pass uniformly over all the surfaces of the rotor and casing, a number of small steam-supply inlets are fitted equidistant in a ring round the covers of the casings at the steam supply end to ensure a uniform distribution of the steam for this purpose.

These inlets on the astern cylinders are also used whilst the engines are under way. to ensure these cylinders being kept sufficiently hot as to be efficient for reversing in case of emergency. The steam supplies to glands are also used for warming-up purposes when preparing for steaming.

The principal advantages of turbine machinery for warships over reciprocating engines are :—

A saving in weight principally due to a decrease of about 15 to 20 per cent. of the number of boilers required for full power, owing to the increased economy of turbine machinery at the highest power. There is also a small saving on the engine weights.

Reduction in the number of working parts and greater simplicity in the machinery installation generally.

Less vibration and less liability to break down on service owing to the elimination of heavy reciprocating parts.

Ease of manipulation in manœuvring the engines.

The steam cylinders can be placed lower in the vessel, which leads to increased protection.

The reduction in numbers of boilers causes a corresponding reduction in stoker complement, in addition to which there is a slight reduction of the numbers required in the engine rooms.

Less oil obtaining access to internal parts and consequently less liability of injury to boilers by admission of oil with feed water. Propellers less liable to race in a sea-way, owing to the smaller diameter and greater immersion.

Less expenditure of stores.

Disadvantages.—The disadvantage originally experienced with turbine installations as regards manœuvring the ship has been overcome by increasing the efficiency, and hence the comparative power of the astern going turbines, and by supplying astern turbines to all the shafts, which is effected by fitting the astern turbines in series.

In the two-shaft arrangements with Parsons or Brown Curtis combined impulse and reaction turbines an astern turbine is in the same casing as each ahead turbine. The power for going astern is however still less than with the reciprocating engines.

Other disadvantages of turbine engines are the considerable falling off in economy at low powers and the large and heavy machinery parts required to be lifted and transported in the engine room when carrying out examinations or repairs.

CHAPTER XXII.

PROPULSION.

Resistance of ships.—In considering the question of the propulsion of ships, it will be necessary in the first place to explain briefly the genera. nature of the resistance experienced by ships in their passage through the water, which resistance has to be overcome by the action of the propeller. The most important element is the frictional action of the water itself on the skin of the ship. Water is not a perfect fluid, and when it is disturbed its particles will exercise friction, both on each other and also on the surface of any body past which they move. If a well-formed ship with a clean bottom be towed through the water, the water will open out at the bow, and follow round the sides of the ship in well-defined currents, called '*stream lines*,' closing in again under the stern, so that the counter pressure under the stern thus caused tends to balance the head resistance to the ship's motion.

If the run aft be not sufficiently fine to allow the water to close in under the stern properly, an eddying wake would be formed under it, which would increase the direct or head resistance to the motion of the vessel. The surface or wave-making action constitutes another source of resistance.

Elements of total resistance.—The three elements constituting the total resistance to the ship's motion are, therefore :—

1. Frictional resistance, due to the gliding of the particles of water over the rough skin of the ship.
2. Eddy-making resistance, due to a wake at the stern.
3. Surface disturbance or wave-making resistance.

The first of these is in general the most important. The second is small and, as it is difficult to dissociate it from the wave-making resistance, it is considered separately in exceptional cases only, and the eddy and wave-making resistances are often grouped together for convenience and considered as the 'residuary resistance.'

Summary of principal facts.—Sir William H. White, in his 'Manual of Naval Architecture,' gives the following summary of the principal facts relative to the total resistance offered to the motion of a ship when towed, or propelled by sails, through the water. The effect of the action of the propeller in increasing the resistance will be pointed out further on.

1. 'That *frictional resistance*, depending upon the immersed surface of a ship, its degree of roughness, its length, and (about) the square of the speed, is not sensibly affected by the forms and proportions of ships, unless there be some unwonted singularity of form or want of fairness. For *moderate* speeds, this element of resistance is by far the most important; for *high* speeds it also occupies an important position—from 45 to 60 per cent. of the whole resistance, probably, in a very large number of classes when the bottoms are clean; and a larger percentage when the bottoms become foul.'

2. 'That *eddy-making resistance* is usually small, except in special cases, and amounts to some 8 or 10 per cent. of the frictional resistance. A defective form of stern may cause largely increased eddy-making.'

3. 'That *wave-making resistance* is the element of the total resistance which is most influenced by the forms and proportions of ships. Its ratio to the frictional resistance, as well as its absolute magnitude, depend upon many circumstances, the most important being the forms and lengths of the entrance and run in relation to the intended full speed of the ship. For every ship there is a limit of speed, beyond which each small increase in speed is attended by a disproportionate increase in resistance; and this limit is fixed by the lengths of the entrance and run—the "wave-making features of a ship."'

Frictional resistance.—Frictional resistance varies with the amount of the immersed surface of the ship, with the co-efficient of friction of the skin in water, and also depends upon the length of the surface, and the velocity with which the particles of water glide over it. The length has an important influence. For example, in some experiments made by the late Mr. Froude, it was found that a plane 8 feet long coated with varnish, moving at a speed of 600 feet per minute, experienced a resistance of 0·325 lb. per square foot, whereas the resistance opposed to a similar plane 50 feet long moving at the same speed was only 0·25 lb. per square foot, or one-fifth less.

For a greater length than 100 feet it has been found that there is still a further decrease in the frictional force per square foot. For lengths up to 300 or 400 feet, the resistance per square foot is slightly less than the case of the 100 foot length. At greater lengths of 600 feet or so, the further decrease is not very marked. The skin resistance of a ship may be regarded as practically the same as that of a plane of the same area as the immersed surface of the ship, with similar ratios of length to depth. The resistance was found to vary as the area and condition of the wetted surface and as the speed raised to the power of 1·83.

Eddy and wave-making resistance.—This cause of resistance may be reduced to a minimum by making the lengths of entrance and run of the ship sufficiently great for her required maximum speed; for a certain speed of a ship there are minimum lengths of entrance and run which must be given in order to reduce the loss from surface disturbance to reasonable limits. This condition is more affected by the length of run than by length of entrance; and the entrance may be reduced to a certain extent without entailing so great a loss of efficiency as would result from a similar decrease in the length of the run.

For every ship there is a certain limit of speed, beyond which any addition can only be obtained at the expense of a very rapid increase in the resistance; and this is attributed to the wave-making action.

The reasons for this are best illustrated by a brief consideration of the various wave actions. The length between two waves of a series depends principally on the speed at which they travel, the length varying with the square of the speed.

From the bows and sterns of all ships in motion, waves are formed, to a greater or less extent, which in most cases pass away from the ship in divergent directions; and the energy expended in creating these waves is wasted. The resistance caused by the maintenance of the diverging wave system is of relatively great importance only at low speeds, and the consideration of the transverse wave system is in general of more importance. The crests of these waves are normal to the direction of the ship's motion. They appear near the bow, rise and fall along the ship's side and, after mingling with a second similar set produced near the stern, they eventually vanish astern of the ship. The lengths of these transverse systems correspond to the speed of the ship, and their combined effect on the ship's resistance depends on the relative position of the entrance and run of the ship which set up the bow and stern system respectively. At low speeds the effect of these waves also is unimportant, but as the speed is increased the total resistance fluctuates about a gradually increasing value, until a critical speed is reached when the two systems combined set up a resultant large system, and beyond this speed the resistance increases rapidly.

In designing a ship, therefore, a suitable length must be chosen which will keep the loss due to surface disturbance within reasonable limits. In small high-speed craft the wave-making resistances are, however, of primary importance. The depth of water also influences the length of the surface waves and, consequently, the resistance of the ship, and, as pointed out by Sir Philip Watts before the Institution of Naval Architects, in certain depths of water a critical speed is reached with the expenditure of a certain power, and a further increase of speed can be attained by the expenditure of actually less horse-power in certain circumstances of speed and depth of water.

Total resistance.—The sum of these three elements constitutes the total resistance offered by the water to the motion of a ship towed through it, when the depth of water is great in proportion to the speed. In a steam-ship there is also an augmentation of resistance due to the action of the propellers.

For purposes of illustration it may be assumed that the total resistance varies as the square of the speed, and it is found, with this reservation, that at the same speed the total resistance in two similar ships varies as the square of their dimensions. The displacements vary as the cube of the dimensions, and it follows that the ratio of the horse-power required per ton of displacement decreases as the size of the ship is increased, i.e., for economical propulsion a large ship is better than a small one, this being particularly apparent when high speeds are required.

Effect of wind and waves.—The laws of resistance given above relate only to smooth water, and do not take any account of the action of the wind and waves, which action, with the resultant pitching and tossing, will evidently cause the resistance of a ship in a seaway to be very different from that in smooth water, and it is impossible to make a theoretical estimate of the difference. It is, however, clear from general observation and experience that length, size, and weight in ships tend to give them greater facilities for maintaining their speed in a seaway, and this is conclusively shown by the regularity with which large ocean steamers make their voyages under all conditions of wind and sea.

In order to make a complete investigation of the theory of propulsion, so far as it has yet been developed, it would be necessary to employ somewhat extensive mathematical reasoning which would be beyond the province of this treatise, so we shall confine ourselves to summarising, with only slight use of mathematical expressions, the leading principles and deductions that illustrate the action of a propeller in the water.

Principles of momentum and work.—The action of propellers is best analysed by the principle of momentum, by which the effect of a force is estimated by multiplying it by the *time during which it acts* instead of the *space through which it acts*, as in the principle of work.

If a body move from rest in a straight line under the action of a constant force P, then after t seconds

$$P\,t = \frac{W}{g}\,v,$$

where W = the weight of the body,
v = the velocity in feet per second,
and g = the accelerating force of gravity.

The product, $\frac{W}{g}v$, is called the momentum of the body; so that the force multiplied by the time through which it acts is equal to the momentum of the body; if the body had initially a given velocity, 'change of momentum' should be substituted for momentum.

If $t = 1$, then $P = \frac{W}{g}v$, that is, *the force acting is equal to the momentum or change of momentum generated per second.*

If in the time t the body has moved through a space x, we have also

$$P\,x = \frac{Wv^2}{2\,g}.$$

Now, $\frac{Wv^2}{2\,g}$ is the kinetic energy of the body, so that the force multiplied by the *distance* through which it acts (or the work done) is equal to the kinetic energy generated; if the body had initially a given velocity the work done would be equal to the change in the kinetic energy.

Therefore, in the simple case of a constant force acting on a body, in a given direction, if it be considered by the *time* during which it acts, its measure is the momentum or change of momentum produced; whilst if it be considered with respect to the distance through which it acts, it should be estimated by the change produced in the kinetic energy.

The general action of propellers can best be understood by considering, first, a few elementary examples.

Pressure of a jet on a fixed plane.—In the first place, by the application of the principle of momentum, the pressure produced on a fixed plane by the impact of a jet of water striking it perpendicularly, as shown in Fig. 278, is easily ascertained. Any particle is at first moving with a velocity v, perpendicular to the plane, and after a certain time it is deflected, and moves parallel to the surface of the plane, so that the momentum, perpendicular to the plane, becomes zero, being destroyed by the action of the plane. This is on the assumption that nothing in the nature of a rebound occurs, and that all the motion perpendicular to the plane is destroyed. This is true for all the particles in the jet, so that the effect of the plane is to destroy the whole momentum of the jet, which originally had a uniform velocity v perpendicular to the plane.

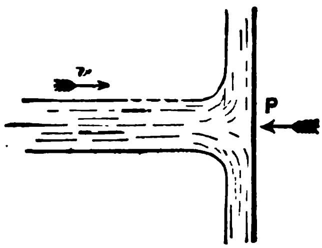


FIG. 278.

If $A =$ area of the jet in square feet, v its velocity in feet per second, and W the weight of a cubic foot of water, $W A v =$ weight of water that strikes the plane per second.

The original momentum of the water delivered by the jet, in one second, perpendicular to the plane was

$$= \frac{W}{g} A v \times v = \frac{W}{g} A v^2.$$

This is entirely destroyed by the action of the plane, and consequently the pressure on the plane will be

$$P = \frac{W}{g} A v^2.$$

It is evident that the greater the quantity of water acted on per second, the greater will be the pressure produced; and this is equally true with respect to the action of propellers.

When a vessel is propelled by the action of any propeller worked by forces within the ship herself, the total momentum of the sea and ship is necessarily zero, and any forward momentum generated by the passage of the ship is exactly balanced by the backward momentum generated by the propeller. We will now proceed to consider actual propellers, commencing with a definition.

The 'race of a propeller' is the technical name given to the stream of water driven sternward by the propeller.

Jet-propeller.—This is the simplest form of propeller, but one that has been rarely used, for reasons that we shall see. In this case, water is drawn from the sea by a pump through orifices in the bottom, and projected sternwards through orifices either at the sides or stern. The water acted on was originally at rest. It is drawn into the ship, and consequently caused to move forward with a velocity V = the speed of the ship, and is lastly projected sternward with a velocity v, which we will suppose to be known.

The final velocity of the issuing water with respect to still water is therefore $= (v - V)$, and if A = the joint sectional area of the orifices, in square feet, the number of cubic feet of water acted on per second is $= Av$. The weight of the water leaving the ship per second is therefore $WA\,v$, and the total sternward momentum per second of the issuing jets is

$$= \frac{W}{g} Av\,(v - V).$$

From the principles previously stated, this must be equal to the reaction produced, or the *thrust of the propeller.*

If R be the thrust of the propeller,

$$R = \frac{W}{g} Av\,(v - V) \quad . \quad . \quad . \quad . \quad (1)$$

For sea-water $\frac{W}{g}$ is equal to 2 approximately.

It is evident that the thrust of a jet-propeller is theoretically independent of the position of the orifice, whether above or below the water level.

Theoretical efficiency of the jet-propeller.—The efficiency of the jet-propeller, if the loss from friction of the passages, shock, &c., be neglected, can now be ascertained. The work done in forcing out the jets of water is equal to the kinetic energy generated; or

$$= \frac{W}{2g} A\,v\,(v - V)^2.$$

The useful work done in propelling the ship $= R\,V$.

$$\therefore \text{ Total work done per second} = R\,V + \frac{W}{2g} A\,v\,(v - V)^2$$

$$= \frac{W}{2g} A\,v\,(v^2 - V^2) \text{ from (1).}$$

The efficiency of the propeller $= \frac{\text{useful work}}{\text{total work}}$.

$$= \frac{R\,V}{\frac{W}{2g} A\,v\,(v^2 - V^2)} = \frac{\frac{W}{g} A\,v\,(v - V)\,V}{\frac{W}{2g} A\,v\,(v^2 - V^2)} = \frac{2\,V}{v + V}$$

This is theoretically the maximum efficiency that any propeller can attain, as it is assumed that all the water is projected directly sternwards, and all the losses from friction of passages, shock, &c., are

neglected, conditions which are far from being even nearly obtained in practice. It is assumed, for example, that the water enters the orifices at speed V, and that its velocity is *gradually* accelerated up to the speed v, so that all the energy in the supply-water to the pump is utilised. In practice much of this is lost, and in some designs practically all. The smaller is v for a given value of V, the greater is the efficiency.

From the formula (1) we see that the thrust mainly depends on the product A v, and that the smaller the value of v the greater must be the value of A for the same thrust. Since the efficiency becomes a maximum when $v = V$, which is the smallest value v can have, it follows that theoretically the larger A is made the more efficient would be the performance. Generally, A is made as large as practical considerations will admit, so as to keep v, the speed of the race, as small as possible; the sternward momentum of the race with respect to still water representing a loss, and the higher its velocity the greater will be the loss from shock, &c. With reference to screw-propellers, as we shall see in a later portion of this chapter, this principle requires modification.

Advantages and disadvantages of water-jet propulsion.—The advantages claimed for water-jet propulsion in warships consist in the freedom from damage in action by wreckage or grounding, greater control of motion of the ship from deck without altering the motion of the engines, and possession of large pumping power in case of a leak.

In practice they are at a great disadvantage, owing to the magnitude of the frictional resistances and the difficulty of operating on a sufficiently large body of water on this plan; and instead of being more efficient than other propellers, as they should be theoretically, they are in practice much less efficient. Their defective action therefore, due to the resistance of passages, &c., combined with the practical objections to the fitting of large orifices in the ship's side, either above or below the water, places the jet out of the region of practical propellers, except under very special circumstances, such as in lifeboats, &c.

Results of trials.—The 'Waterwitch' is the only example of a ship with a jet-propeller in the Royal Navy, and her trials demonstrated the inefficiency of the system. With 760 I.H.P. the 'Waterwitch' attained a speed of 9·3 knots, displacement 1,160 tons. The 'Viper,' twin-screw gun-vessel, of somewhat similar dimensions but inferior in form, attained a speed of 9·6 knots with 696 I.H.P., displacement 1,180 tons. The quantities of water acted on by the two kinds of propellers were very different. In the 'Waterwitch' the quantity of water passing astern per second was about 150 cubic feet, while the twin screws of the 'Viper' acted on over 2,000 cubic feet per second, or about fourteen times as much.

In 1883 one of the second-class torpedo boats by Messrs. Thornycroft & Co. was fitted with a turbine, and great skill and care were exercised to insure the best results. Efficiency of astern working was subordinated to that of ahead working, and the water inlet was placed at the bottom, and made into a scoop to utilise as much as possible of the energy of the entering water. All sudden changes of angles and velocity of water were avoided. On the comparative trials between the hydraulic boat and the other boats of equal size fitted with screw-

propellers, it was found that the speed of the hydraulic boat was no greater than could be attained in the screw boats with about half the power. The actual results were: Hydraulic boat, I.H.P. 167, speed 12·6 knots. Screw boat, I.H.P. 170, speed 17·3 knots. The efficiencies were analysed as follows: Screw boat: Engine, ·77; screw propeller, ·65; total efficiency, ·5. Hydraulic boat: Engine, ·77; jet-propeller, ·71; circulating pump, ·46; total efficiency, ·254. The screw boat was therefore nearly double as efficient as the hydraulic, the principal loss in the latter being in the pump.

Feathering paddle-wheels.—With feathering paddle-wheels the floats are supposed to act in a direct sternward direction, and to enter and leave the water normally, the area of the race of both paddles being equal to that of a pair of floats. In ordinary radial paddle-wheels there is much local agitation and disturbance of the water, due to their oblique action on entering and leaving, which complicates the question, and the area of the race is not so clearly defined.

Before being acted on by the floats the water is assumed to be at rest, and therefore, relatively to the ship, it would have a sternward velocity V, equal to the speed of the ship.

Let v be the final velocity of the race *relatively* to the ship, and $A =$ the area of a pair of floats, one on each side of the ship, then the sternward momentum generated per second is equal to R, the propelling reaction, or $R = \frac{W}{g} A\, v\,(v - V)$, as in the jet-propeller.

The propelling reaction R acts on the paddle floats, and the velocity of the floats is assumed to be v, the same as that of the propeller race relatively to the ship; therefore the engine has to overcome a resistance R through a space v in one second. Hence, as regards the efficiency of the paddles the energy exerted in propelling is $= R\,v$, and the useful work done is clearly $= R\,V$.

Therefore the total work wasted is equal to

$$R\,(v - V),$$

and the efficiency is

$$\frac{R\,V}{R\,v} \text{ or } \frac{V}{v},$$

which is theoretically not so great as in the jet-propeller, other things being equal.

The work wasted in producing the race is equal to the kinetic energy generated

$$= \frac{W}{2g} A\, v\,(v - V)^2 = \tfrac{1}{2} R\,(v - V).$$

This, therefore, only amounts to one-half the total power wasted, the remainder being absorbed in producing the violent churning and agitation of the water which is always produced by paddle-wheels. In practice the loss from this cause would be even more than one-half of the total power wasted, for in the above investigation we have neglected the resistance to forcing the floats in and out of the water, which considerably increases the work of the engine.

The expression $\frac{V}{v}$ must therefore be regarded as the maximum possible efficiency with these propellers.

Radial paddle-wheels.—In paddle-wheels with radial floats only the float for the time at the bottom of the wheel is vertical, and giving direct sternward velocity to the water. All the others act obliquely and have a vertical as well as a horizontal reaction, the former, although absorbing a large proportion of the power of the engines, being wasted, so far as propulsive effect is concerned. The greater the immersion of the wheel, the greater will be the loss due to the vertical component of the pressure on the float. These wheels, therefore, should be so designed that at the maximum load draught of the ship, they should not be immersed more than one-quarter the diameter of the wheel; for beyond this limit the loss from the vertical reaction increases at a very rapid rate.

With these propellers it is impossible to determine, with accuracy, the area and speed of the race. Various assumptions may be made, but they are now of little practical value, as the use of the feathering paddle is almost universal for important vessels in which this means of propulsion is employed.

Objections to paddle-wheels.—The chief objection to the employment of paddle-wheels for ocean navigation arises from the practical difficulties attending the variation in immersion during long voyages, owing to the lightening of the ship by the consumption of coal, stores, &c. Even with feathering wheels, in which the floats are approximately vertical when in the water, the loss from forcing the floats in and out of the water, churning, &c., is much increased when the wheels are deeply immersed; and it is evident that if the wheels are to be sufficiently immersed at the end of a long voyage they must have been too deep in the water on starting. Even if water ballast be used to overcome this difficulty a larger expenditure of engine power would be necessary. For short voyages, in which the draught of water is comparatively unchanged, paddle-wheels may be advantageously employed, and they are almost essential for propulsion in many shallow rivers, where the depth of water is insufficient to admit the use of screws.

For ocean navigation, however, they have been superseded by the screw, for in addition to the loss of efficiency from alteration in draught, paddle-wheels are objectionable in consequence of the racing and straining of the machinery due to the rolling motion in a seaway, causing the paddles to often emerge from the water on one side and be correspondingly depressed on the other. For vessels of war the exposure of the paddles and engines to injury by shot and their interference with the deck arrangements render them doubly inadmissible. The paddle-wheel, as a propeller, is now only employed in the few special cases suitable to this form of propeller.

Screw-propeller.—The action of a screw-propeller is much more complex than that of the two types of propellers previously discussed, and is due mainly to the following causes:—

1. The action of the propeller on the water is oblique instead of direct.

2. The velocities of the several particles of water acted on by the screw are different from each other.

3. The screw acts on water that has previously been set in motion by the ship.

The difficulties attending an exact mathematical investigation of

the action of screw-propellers are consequently so great that the problem is not yet solved. No formula yet in existence will give an accurate value for the thrust of an ordinary screw, even when working in undisturbed water, while for the case of propellers working behind actual ships the problem is still more difficult and complicated. The principles involved may, however, be easily understood. Oblique action, other things being equal, is always a cause of loss of efficiency in a propeller, and the fact that, in spite of this, the screw is a practically efficient propeller is explained by the circumstance that it operates upon a much greater quantity of water than could be acted on by a pair of paddle floats, or by any other propeller in a ship of the same size in the same time, and, as has been proved previously, the efficiency of any propeller depends to a large extent on the quantity of water acted on. The screw is the form of propeller best adapted to fulfil this condition.

It will now be desirable to define a few of the technical terms that will frequently be used.

Diameter.—The diameter of the screw is the diameter of the circle formed by the tips of the blades when revolving. The area of this circle is called the 'disc area' of the screw.

Pitch.—The pitch of the screw is the distance through which the screw would advance *in one revolution* provided it revolved in an unyielding medium such as a solid nut.

Speed of screw.—The speed of the screw is the distance it would advance *in a unit of time*, supposing the screw to be working in a solid nut. This is obviously equal to the pitch of the screw multiplied by the number of revolutions made per unit of time.

Slip.—In consequence of the screw-propeller working in a yielding medium, the speed of the ship is generally less than the speed of the screw. The difference between the speed of the screw and the speed of the ship is called the slip of the screw.

If v = speed of the screw.

V = " " ship.

$v - V$ = slip of the screw.

$\frac{v - V}{v}$ = slip of the screw expressed as a fraction of the speed of the screw . . . (2)

$\frac{v - V}{v} \times 100$ = percentage of slip.

This, however, is only the *apparent* slip of the screw. It assumes the screw to be acting on water previously at rest, which can never be the case with the water operated on by the screw-propeller of a ship. The friction of the ship on the water during its passage causes a wake to follow her, so that the screw-propeller acts on water already set in motion. The velocity of this stream must therefore be considered, in order to obtain the *real slip*, which represents the true value of the backward velocity impressed on the water by the propeller. The speed at which the water follows the ship depends on her form, and is difficult to ascertain, so that the slip generally referred to is the apparent slip only and not the real slip.

If we assume the water to be a stream of velocity μ and of sufficient breadth, the original velocity of the water acted on, relatively

to the screw, is $V - \mu$, while its final velocity is v, so that the *real slip* will be $\frac{v - (V - \mu)}{v}$

$$\text{or} \qquad \frac{v + \mu - V}{v} = \text{real slip} \quad . \quad . \quad . \quad (3).$$

Negative apparent slip.—From the nature of the medium in which the screw-propeller works, it is clear that in every case there must necessarily be *positive real slip*, as it is the change in the backward momentum of the water which causes the propelling reaction or thrust; but it is not difficult to imagine cases in which the water following the ship has such an initial velocity that the *apparent slip* might be *negative*—that is, the speed of the ship might be greater than that of the screw-propeller. From the formula (3) above, it will be seen that if μ be considerable, V may be greater than v quite consistently with a positive value for the real slip, in which case the apparent slip would be negative. This has sometimes occurred, and before the matter was sufficiently understood gave rise to a vast number of theories to account for its existence.

Possibly in some cases the apparent negative slip may be attributed to some extent to the difficulty of correctly estimating the true mean pitch of an ordinary screw. The method adopted practically is to divide each blade by a number of circular arcs at equal distances apart, as in Fig. 300, and to measure the pitch or pitches at each arc séparately. It is sometimes found that the pitches at the various radii are somewhat different; and frequently at each radius the pitch of the leading part differs from that of the following part of the blade. The arithmetical mean of all the pitches thus measured, for all the screw-blades, is called the 'mean pitch' of the screw; but considering the different velocities with which the several sections pass through the water, and their different obliquities and areas, it is by no means certain that this method gives the true mean pitch of the propeller as regards its propulsive effect, and a comparatively small error in estimating the mean pitch might considerably affect the calculated apparent slip of the screw.

Propeller race.—The action of the screw-propeller is to drive sternward a cylindrical column of water, usually called the 'propeller race,' and the thrust of the screw is measured by the sternward momentum generated in this race in a unit of time. The area of this race is approximately equal to that of the screw disc, less that of the boss of the screw, the race being in fact approximately a revolving annular column. In consequence of the obliquity of the propelling surfaces, the race receives a rotatory as well as a sternward motion, and this centrifugal action causes a certain loss of thrust. The race of a screw-propeller may be conceived to be a series of concentric cylinders of water moving sternward, and rotating at different velocities. It is evident that the thrust must be diminished both by the centrifugal motion and by the frictional action of the particles of water.

Augmentation of resistance due to action of screw-propeller.—The most important feature in the action of a screw-propeller, as affecting its efficiency, is the effect it produces on the water under the

stern of the ship. In the absence of the propeller the water displaced at the bow by the passage of a well formed ship, would close in under the stern and cause a forward pressure there. The action of the screw withdraws this water, and consequently diminishes the pressure of water under the stern, which is equivalent to increasing the resistance of the ship, as compared with the natural resistance, or the resistance experienced by the ship when towed at the same speed. The *resistance augmentation* varies exceedingly, depending on the shape of the stern of the ship and the size and position of the screws relatively to the vessel. The late Mr. Froude experimented, and considered about 40 per cent. to be the augmentation for a single-screw ship with a full run, and thick stern- and rudder-posts, a large percentage being accounted for by these posts alone. Later experiments have indicated for single-screw ships from 20 to 40 per cent., depending on the form, and the stern- and rudder-posts; the lower value would be for a very fine ship. For twin-screw ships it also varies considerably, say from 5 to 25 per cent. The smaller value is, as before, for vessels of very fine form and usual position of screw. In the 'Iris,' and other twin-screw ships of similar form, the increase is 10 to 12 per cent. It may be much reduced by placing the screw some distance behind the ship.

Mr. Froude proved by experiment that if a single screw were placed, from one-third to one-quarter of the extreme breadth of the ship, clear from the stern, the increase of resistance due to its action was only one-fifth of that ordinarily produced. Even omitting the practical objections to such a position, it must not be thought, however, that such a change would be entirely beneficial, for although the ship's resistance would be reduced, yet the propeller would be further removed from the following wake of water, so that it is less able to utilise the energy existing in this wake. The initial velocity of the following wake causes the thrust of the propeller to be greater than if the water were undisturbed.

General conclusions.—Notwithstanding these defects the screw-propeller has proved itself practically to be the most efficient propeller. It has entirely superseded the paddle-wheel for ocean navigation, as it is very slightly affected by causes which have such a prejudicial effect on the efficiency of paddle-wheels—viz. variation of immersion and rolling in a seaway—and is protected from shot by being below the water.

The considerations respecting screw-propellers indicate that the greater the area the greater will be the efficiency. This is generally true, but is subject to modification in practice. It is most important that the highest part of the screw-blades should be a sufficient depth below water to prevent air to any considerable extent mixing with the propeller race, by being drawn down or breaking the surface of the water, which would decrease the quantity of water acted on by the screw. For so long as air does not obtain access to the blades, the full atmospheric pressure is available for causing water to flow to the propeller, but when this is reduced by access of air the flow is diminished, and efficiency lessened. As the lowest point of the screw-blade must, in ordinary ships, be above the keel, these considerations limit the maximum diameter that can be advantageously given to the screw. If, however, the blade area be too small, and the thrust too concentrated,

an action termed 'cavitation' occurs, the water being unable to follow up the motion at the back of the blade, cavities are formed in the column of water being acted on, and loss of efficiency results.

With screw-propellers revolving rapidly it is evident that there must be a considerable waste of power in overcoming the edgewise and frictional resistance offered by the water to the motion of the screw. The power expended in this work is in many slow-moving engines estimated at about 4 per cent. of the total I.H.P. developed by the machinery, while with fast-running engines it will exceed this. In considering the actual method of working of a screw-propeller the slip of the screw must not be regarded as being merely a loss, as it is by virtue of the slip that the propeller derives its thrust. From theoretical considerations the screw, if working ahead with no slip, would experience a backward thrust due to this frictional resistance, but from actual experiments it is found that the recorded thrust is then sensibly zero. This again points to the difficulty in estimating the real pitch of the screw, as it would appear that the curved sections cause eddies on the forward face or back of the blade, which relieves it of pressure sufficient to neutralise the backward thrust due to these frictional forces. The frictional resistance of the circumferential part of a large screw, moving at a high velocity, is much greater than that of the part nearer the boss, and this modifies in practice the proportions arrived at from theoretical considerations. The disc area in ordinary screw ships varies from one-half to one-quarter of the immersed midship section of the ship, one-third being a good average value.

The most important consideration relative to the efficiency of a screw-propeller is the facility offered for the unrestricted flow of a plentiful supply of water to be operated on, and this is attained by making the after run of the ship with as fine lines as possible, but was not fully understood when screw-propellers were first introduced.

Guide-blade propeller.—Mr. Rigg proposed a screw having fixed 'guide blades,' placed immediately behind the revolving propeller, so arranged that the rotating water leaving the revolving part is discharged on to the fixed guide-blades, which are inclined at the reverse angle to the blades of the screw, so that the rotation of the water is gradually destroyed, and the water caused to be moving directly astern on leaving the guide-blades. A thrust is therefore exerted on the guide-blades which is added to the propulsive power.

Screw-turbine propeller.—Mr. Thornycroft in his 'screw-turbine' propeller, Fig. 279, has adopted these guide-blades, and added other features having for their object the gradual acceleration of the motion of the water by the screw. In the ordinary propeller the leading edge acts suddenly on the water it meets, and, as we have seen, such sudden action causes loss. In Thornycroft's screw-turbine the screw is made to revolve in a tunnel, A, and the boss of the screw is of increasing diameter from forward to aft, so that the area for the passage of water inside the tunnel is gradually reduced. The water enters the tunnel with the velocity of the vessel, and passing through the diminishing area gradually increases its velocity. The screw is of considerably increasing pitch to suit this gradual increase of velocity, and at the after end are the fixed guide-blades, with a long tail to allow the water to gradually return to the velocity corresponding to the larger area.

The thrust on the fixed guide-blades is considerable, and the whole device forms a propeller, which is able to act on a smaller quantity of water than an ordinary one. It is thus suited to shallow draft vessels, and many steamers for operating on the Nile have been so made.

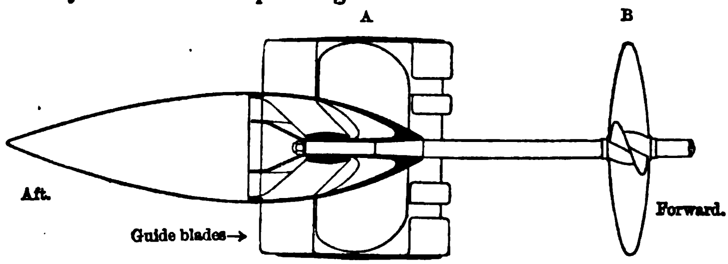


Fig. 279.

This propeller is, however, very inefficient when going astern, so that for astern working a second ordinary propeller, B, is fitted.

Messrs. Yarrow, in similar boats, obtain a light draught by rotating the propellers in a tunnel in the stern, so that the top of the screw may be considerably out of water. The action of the propellers fills the tunnel, so that the former are fully supplied with water.

Twin screws have been employed in the Royal Navy for many years with good results. The system has also been applied to passenger steamers, and is gaining favour in that service. The area of the propeller race has by this means been much increased, and trials of single and twin-screw ships show that the propulsive efficiency of twin screws is rather greater than that of single screws.

The duplication of the machinery in twin-screw ships prevents a total collapse in case of accident; for if one set of engines broke down, the other set would be capable of propelling the ship at a fair rate of speed. By the use of twin screws the manœuvring power of the ship is greatly increased, for by working one screw ahead and the other astern the ship may be turned in her own length without necessitating any speed of ship, and this may be of great importance in some cases. The ship could also be steered by the propellers alone, without the rudder, in case of accident. In a twin-screw ship with vertical engines the hull can be subdivided into longitudinal compartments, by fore-and-aft bulkheads, which add to the strength and safety of the ship.

Low power working with twin screws.—It was at one time thought that with twin screws the most economical method of working at low powers was to stop one engine in order to get rid of its friction, and develop all the power in the remaining engine, using helm as necessary to maintain a straight course. Comparative trials have, however, been made in the Royal Navy, with the result that the most economical method of working was shown to be when using both screws, although the power developed in each engine may become very small.

Triple screws have not been fitted in the Royal Navy for reciprocating engines, but various war vessels in France, Germany, and the United States, several of which have been recently tried, have been provided with triple screws. On this plan three independent sets of engines are fitted in separate watertight compartments working three separate

screws, one at the centre line, and one at either wing. The engines of the centre screw are placed abaft those of the wing screws, those for the two wing screws being arranged as in a twin-screw vessel. By this means, with large powers, the engines become of more moderate size, and more easily stowed; but the principal reason for its adoption is that at low powers the centre engine only can be used, so that it will be of moderate size for the power developed. It will therefore, as compared with the twin-screw system, be more economical in fuel *per I. H. P.* at low powers, while the constant friction of the engines at work will be much reduced. The drag of the two idle screws adds to the resistance of the vessel, so that it is possible the consumption of fuel for a given distance steamed by the vessel at low powers may not be less or may even be increased. Experience alone will decide whether this system is on the whole advantageous for warships. Triple screws are being fitted to many mercantile ships and some British warships in connection with turbine machinery, as there are advantages as regards machinery arrangements obtained with it, although the engine-room is not subdivided by a watertight bulkhead.

Quadruple screws.—The recent British warships have quadruple screws, the two shafts on each side of the ship being driven by independent sets of turbines in two or three watertight compartments.

The efficiency of the propeller is intimately associated with (*a*) the shape and general design of the hull; (*b*) the position of the propeller with relation to the hull; (*c*) the displacement and speed of the ship and the dimensions of the propellers. Owing to the friction between the skin of the vessel and the water, the propeller works in a current moving in the same direction as the vessel, and this current or 'wake' increases the propeller efficiency. If the vessel were not propelled by a screw, the whole of the energy of the wake would be lost, but the screw propeller turns a certain amount of the energy of the wake into useful work, thus increasing the efficiency. However, the whole of the energy of the frictional wake has to be provided by the propelling machinery, and it is, at moderate speeds, the principal source of loss, even allowing for what is regained by the action of the screw.

The screw propeller being virtually a pump, drawing water from ahead, and delivering the water aft, the reaction due to this is the force propelling the vessel. The energy necessary to overcome the augmented resistance due to the propeller is, for war vessels of ordinary form, approximately equal to the energy the screw recovers from the frictional wake, and these two thus balance one another, and may be considered cancelled in deducing the propeller efficiency.

The resistance the vessel offers to motion through water, and thus the actual *effective horse-power* necessary to be delivered by the propeller, is deduced from testing models in an experimental tank. When the tank experiments are compared with trials with the actual vessel at various powers and speeds, known as progressive trials, and allowance made for the mechanical efficiency by reducing the indicated horse-power accordingly, the efficiency of the propeller is deduced from the ratio between the 'effective horse-power' and the reduced indicated horse-power. For modern war vessels with reciprocating engines its amount is from 60 to 70 per cent., with smaller values in the case of turbine machinery of from 55 to 65 per cent.

CHAPTER XXIII.

CO-EFFICIENTS AND CURVES OF PERFORMANCE.

Co-efficients of performance.—For many years at the Admiralty the following co-efficients have been used to indicate the resistance and propulsive efficiency of ships approximately.

Let A = area of immersed midship section of the ship in square feet.

D = displacement of the ship in tons.

V = speed of ship in knots.

Then,

$$\text{1st co-efficient} = \frac{A\ V^3}{\text{I.H.P.}}$$

$$\text{2nd co-efficient} = \frac{D^{\frac{2}{3}}\ V^3}{\text{I.H.P.}}$$

These are based on the assumption that the resistance offered by the water to the motion of the ship varies as the square of the speed, and that, consequently, the power required to overcome this resistance would vary as the cube of the speed. This is only true for moderate speeds, but the co-efficients calculated from these formulæ have been very useful for comparing the relative performances of ships, especially when somewhat similar in form, and have been valuable, in the case of a new design, as data for approximating to the I.H.P. necessary to drive the ship at an assumed maximum rate of speed. On the whole, the second co-efficient, in which the efficiency is referred to the two-thirds power of the displacement, has been found to be the more trustworthy, giving a fairer measure of the resistance than the midship section co-efficient, especially in dealing with ships that are not similar in form.

It is, however, a well-known fact that at the higher rates of speed the resistance of ships often varies at a very much higher power of the speed than the square, and these co-efficients, though they have done good service, are not now sufficiently accurate, and fail to indicate many points of importance which are shown by more correct methods.

The second coefficient may be used to estimate the approximate change of speed of a particular ship for the same I.H.P., or the change of I.H.P. for the same speed at different draughts within moderate limits. For the same speed, I.H.P. varies as $D^{\frac{2}{3}}$, and for the same horse power, $(\text{speed})^3$ varies as $\frac{1}{D^{\frac{2}{3}}}$. For rough calculations within moderate practical limits the displacement D may be taken as approximately proportional to the draught, but in cases in which the actual displacements for various draughts above and below the ordinary draught is known a close approximation may be made.

Experimental determination of I.H.P.—The power necessary to drive a ship of a new design is now generally obtained from results of model experiments. A model of the ship, generally about 14 feet in length, is made in paraffin wax, and, after being loaded to produce the correct draught, is allowed to float freely in water and constrained to move along the line of keel whilst being towed from a single point by means of a link connected to an overhead truck running on rails, the link being arranged so that it can swing in the direction of the course of the model. The truck is driven by external mechanical power at a series of uniform speeds, and the model, running in the water below, exercises a dragging effect equivalent to its total resistance, which is made manifest by the lagging of the lower part of the connecting link and is measured by the extension of a spring, secured at one end to the lower part of the link and at the other to the fixed part of the truck, the extension being multiplied by linkwork, and automatically recorded on a paper drum, together with time and distance. This gives the resistances of the model at each of a series of speeds, and from these results the resistance, and therefore the power required to drive the full-sized ship at known speeds, can be deduced to a fair degree of accuracy.

The elements of ship's total resistance are dealt with at length on page 287 et seq. ; of these the frictional resistance can be calculated fairly accurately from the formulæ : $R = f\, s\, v^n$, where v is the speed of the ship or model, s the area of wetted surface, and f and n are variable coefficients depending principally upon the nature of the surface and the length of the body.

The sum of the remaining elements, wave-making and eddy resistances, is usually termed 'residuary resistance' and follows what is known as the law of comparison, i.e. the residuary resistance for similar ships running at corresponding speeds varies directly as the displacements. Two ships are similar when they differ only in size, one being say n times as long as the other, n times as broad and n times as deep, while the corresponding speeds are those speeds of the two ships which are in the ratio $\sqrt{n} : 1$.

To obtain the resistance of the ship from the model results, the frictional resistance of the model is first calculated from the above formula for the various speeds, and these results, subtracted from the total resistances of the model at the various speeds, enable a curve of residuary resistance of the model to be drawn on a speed base. In accordance with the law of comparison, this curve represents, on different scales of resistance and speed, the curve of residuary resistance of ship. The frictional resistance of ship is then calculated from the formula, and these results, superimposed on the curve of ship's residuary resistance, give a curve of ship's total resistance on a speed base. From this curve the effective H.P. at each speed is directly obtained, also the I.H.P. by assuming efficiencies for the screw and mechanism.

Curves of I.H.P.—In cases where an accurate analysis is required, trials at several different rates of speed, varying from the maximum to speeds as low as from three to four knots, are carried out, and the I.H.P.s for the respective speeds shown in a graphical form by the construction of diagrams, in which the speeds are set out as abscissæ,

and the corresponding horse-powers as ordinates. Having determined a sufficient number of points, a fair curve is drawn through them, and the I.H.P. corresponding to any intermediate speed can then be ascertained by simply drawing the vertical ordinate from the point representing the speed required.

The details of construction of these curves of I.H.P. are shown in Fig. 280. In this case the speeds at which the ship was tried are those marked A, B, C, and D, and the corresponding horse-powers are shown on the scale of I.H.P. by the points E, F, G, and H. By drawing the dotted horizontal and vertical lines as shown in the diagram, the points P, Q, R, and S are obtained. It is clear that when the horse-power is zero the speed will also disappear; so that if a fair curve be drawn through the points P, Q, R, and S, to touch the horizontal base line at the origin, where the speed is zero, it will represent the

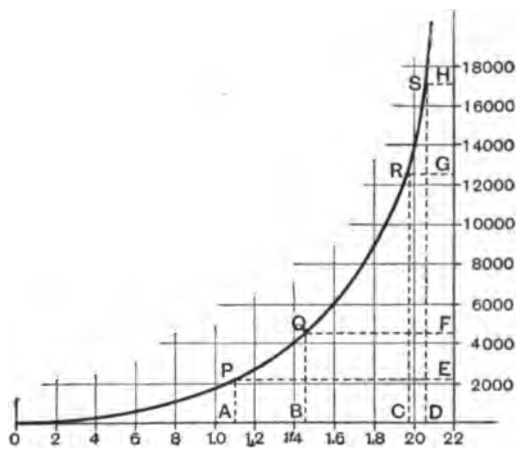


FIG. 280.

varying relations between the I.H.P. and the speed of the ship; or, in ordinary language, the curve of the I.H.P. of the ship. From these diagrams many important facts are learnt.

Curves of indicated thrust.—The late Mr. Froude proposed the substitution of the '*indicated thrust*' of the propeller for the I.H.P. as ordinates of the diagram.

The indicated thrust may be estimated by multiplying the I.H.P. by 33,000 to bring it to foot-pounds, and dividing the product by the speed of the propeller in feet per minute, that is the pitch of the propeller in feet multiplied by the number of revolutions per minute

The indicated thrust is therefore equal to—

$$\frac{33{,}000 \text{ I.H.P.}}{\text{pitch in feet} \times \text{revolutions per minute}}.$$

Having calculated the indicated thrusts for the trial speeds of the ship, a curve was constructed, with the speeds as abscissæ, as in the I.H.P. curves, but with the indicated thrusts as ordinates instead of the I.H.P.s. Fig. 281 is an example showing the construction of an

indicated thrust curve. Such curves have not, however, been found practically useful, as was at one time anticipated.

Components of total power exerted.—The power exerted in the cylinders of a marine engine when analysed can be resolved into several component parts, which show the amount usefully expended and the distribution of the losses.

The following are the components :—

1. Useful thrust, or ship's true resistance, as when being towed.

2. Augment of resistance, due to the action of the propeller in diminishing pressure under the stern of ship, less the gain due to the action of the following wake.

3. Equivalent of friction and resistance of the screw-blades in their motion through the water, and the necessary loss by slip.

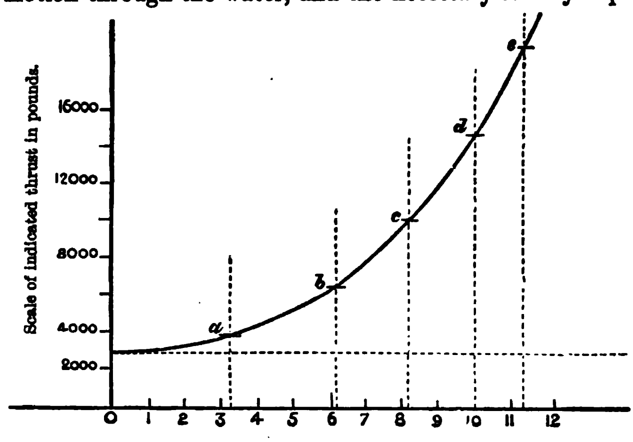


FIG. 281.

4. Equivalent of friction due to the piston packings, glands, and dead weight of the working parts, &c., which constitutes the initial or slow-speed friction of the engines.

5. Equivalent of additional friction of engines due to working load.

6. Equivalent of the duty of any pumps worked off the main engines, such as air-pumps generally, and feed- and bilge-pumps often.

As regards the second element, when a vessel of good form is towed through water the water follows the motion of the ship and closes in at the stern, so that the pressure on the stern is the same as that on the bow. By the action of the screw, however, the stern currents of water are withdrawn, so that the pressure on the stern is diminished, and therefore the resistance to motion is increased. This is a source of loss. Again, the friction of the hull of the vessel causes a body of water, known as the 'frictional wake,' to follow her at a certain speed compared with still water. Now, as compared with the effect of a propeller acting on water previously at rest, the action may be considered to be the same as if a body of water of volume and speed equal to the frictional wake at the propellers were impinging on them, causing a force to be exerted on the screws, and thence on the ship. This represents a gain, and item 2 is the

difference between the loss represented by increased resistance due to the action of the screw, and the gain due to the presence of the frictional wake.

Constant friction of engines.—It was observed by Froude, on drawing the indicated thrust curves, that they do not descend to the thrust zero when the speed disappears, but tend to cross the vertical axis at some distance above it, representing a considerable amount of thrust at the zero of speed. This apparent thrust when the speed is reduced to zero, and when there can be no real thrust, is the thrust-equivalent of the initial or minimum friction of engines and shafting.

Unfortunately this curve cannot be relied on to give accurate results. Large numbers of such curves have been obtained, and a considerable variation is presented in the estimates of initial friction as so ascertained.

Additional friction due to working load.—Besides the initial or constant friction, called sometimes 'dead load friction,' of the engines and shafting—i.e. the frictional resistance of the engine if it were being rotated by an external force with the propeller removed, an additional frictional resistance is caused by the application of the steam pressure necessary when at work.

The total frictional resistances may be conveniently divided into portions—viz. (*a*) the constant friction of the engine and bearings, including the thrust and propeller-shaft bearings; (*b*) the additional friction of the engine and its bearings up to and including the thrust bearing due to the application of the working pressure. Of these frictional resistances (*a*) is practically the same at high as at low powers; the load on the propeller shaft bearings is approximately constant, viz. that due to the weight of the shafting; (*b*) increases somewhat with the power being transmitted, as the load on the engine-bearings and thrust-block depends on the steam-pressures on the pistons, which are, of course, higher at high powers than at low powers. Its amount depends largely on the condition and lubrication of the bearings, and recent investigation indicates that with well-lubricated bearings it is relatively much less at high powers than at low, although exact data are not available.

Determination of mechanical efficiency.—The amount of work lost by friction of the engine-bearings is deduced, at various powers, from records of the indicated work done in the cylinders and the work transmitted by the shaft. The indicated work is obtained from indicator diagrams, whilst the work transmitted by the shaft is known when the mean twisting moment and the speed of rotation have been determined.

For small engines the *mechanical efficiency* (see Chapter II.) is readily calculated from measurements of indicator diagrams, and the readings of a brake or dynamometer applied to the shaft. From the indicator diagrams the I.H.P. is deduced, whilst from the dynamometer readings the brake horse-power is calculated. The ratio $\frac{\text{B.H.P.}}{\text{I.H.P.}}$ is the *mechanical efficiency* of the engine.

Until recently it has been practically impossible to measure the actual twisting moment transmitted by the shaft of a large marine engine. Brakes and dynamometers cannot be applied to engines of

such large powers, and only by the recent introduction of the 'Torsion Indicator' or 'Torsion-meter' (see Chapter XXVIA) has it been possible to make direct observations of the actual twisting moments transmitted by such shafts. The principle of the instrument is to record the angle of torsion between the two ends of a fixed length of the shafting, from which is deduced (either by actual trial in the shop, or by theoretical investigation) the twisting moment causing this angle of torsion. With these instruments it has been possible to record the actual twisting moment transmitted by marine engines of the highest power, from which the mechanical efficiency can be deduced with reasonable accuracy.

The following table shows, for reciprocating engines of certain powers, the approximate ratio which the power transmitted to the propeller shaft, as deduced from torsion-meter records, bears to the indicated horse-power :—

I.H.P. of Engine	Mechanical Efficiency of the Engine (not including propeller shafting) approximately
	Per cent.
1500	88
2000	91
3000	92
5000 and over.	93

To the losses due to bearing friction up to and including the thrust block, must be added the loss due to friction of the propeller shaft bearings. This latter loss is estimated approximately from calculations based on records taken (whilst the vessel is in dry dock) of the moment of resistance to rotation of the propeller shafting when disconnected from the thrust shafting. This moment is found by suspending weights from a propeller blade (at known radius) until the shaft begins to rotate. Two figures can be ascertained in this way—viz., the weight which will start the shaft rotating from rest, and, secondly, the weight which will keep the shaft moving after it has been started. The latter is less than the former and will more nearly represent the friction when at work. Taking the moment deduced from the latter weight we multiply by the speed of rotation at a given power, and this measures the work absorbed by friction of the propeller shaft bearings at that power, assuming the friction when afloat to be approximately the same as when in dock.

Average values of the power absorbed by propeller shaft-frictional losses at full power are about 2 per cent. in battleships and from 2 to 3 per cent. in cruisers. For intermediate powers this loss is approximately proportional to the speed of rotation. Thus we see that, for the latest type of cruiser, about 93 per cent. of the indicated work appears as work transmitted to the propeller shaft, and a further 3 per cent. is wasted in propeller shaft friction ; consequently about 90 per cent. of the indicated work is transmitted to the propeller, and this is the *efficiency of the mechanism.*

The dead-load friction has in many engines been ascertained by actual test. With the horizontal engines of H.M.S. 'Iris' it was shown to represent 8 per cent. of the full power. It is generally from 5 to 10 per cent., depending on the type of machinery, but in small

quick-running engines it is sometimes smaller even than the lower figure. In the 'Minerva' and 'Hyacinth,' both tried with propeller blades removed, it amounted to about 5 per cent.

The relative importance of dead-load friction increases as the power developed is reduced, which partially accounts for the fact that it is not economical to reduce the speed of a vessel beyond a certain point.

Froude's analysis of total power.—From experiments made by Mr. Froude on single-screw ships he concluded that generally in such ships of ordinary form and with slow moving engines such as those in use at that time, with air-pumps, feed-pumps, etc., worked off the main engines, only from 37 to 40 per cent. of the total power exerted by the engines was utilised in useful thrust of the propeller.

Distribution of power in modern vessels.—With a modern high-speed engine the distribution of total power is quite different from the estimate of Froude. In the first place, a continuous reduction of dead-load friction has been effected in recent years, due to improved design and workmanship and to the use of much higher pressures and piston speeds, which has resulted in much smaller engines and shafts for a given power.

The estimates, also, must necessarily vary with the arrangement of pumps at the engines. In some vessels not any pumps are worked off the main engines. In the Royal Navy the air-pumps are the only ones that have recently been worked from the main engines, and these are made considerably smaller than at first, so that the power required for working them is correspondingly small. In recent ships these also are worked by independent engines.

For a naval twin-screw vessel with fast-running, high-pressure engines, and air-pumps worked off the main engines, the distribution of power may be taken to be as follows, approximately :—

	Per cent.
Dead-load friction	6
Working-load friction	7
Air pump working	1
Energy in sternward column of water, loss by blade friction, and augmentation of resistance, allowing for gain due to speed of wake	33
Balance or effective horse-power	53
	100

The sum of the first three items represents the engine losses, and amounts to 14 per cent., corresponding to a mechanical efficiency of ·86. The amount of power delivered to the screw is therefore 86 per cent., also 33 per cent. is lost by the action of the screw before the remainder is transmitted to the thrust block. The efficiency of the screw is

$$\frac{86-33}{86} = 61\cdot6 \text{ per cent.}$$

Recent experiment appears to indicate that with well lubricated bearings the working-load friction may be much less than even the above proportion indicates. In the most recent naval engines the air-pumps are worked independently of the main engines, and these engines being of high speed their friction is correspondingly less, so that with such

an engine the following is an approximation to the distribution of power :—

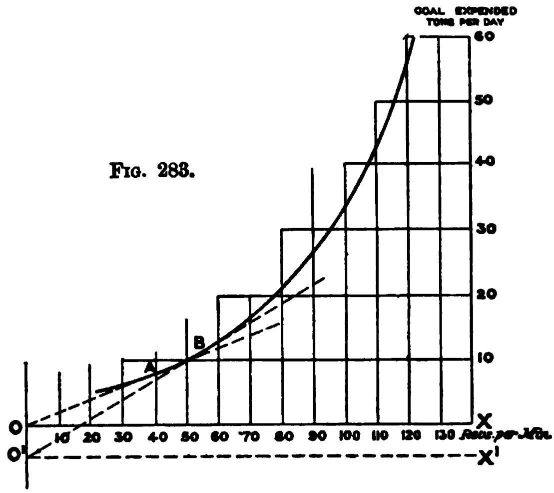
	Per cent.
Dead load and working load friction	11
Loss at propeller, including augmentation and gain due to wake	30
Balance, or effective horse-power	59
	100

The efficiency of the screw in this example is therefore

$$\frac{89-30}{89}=66{\cdot}3 \text{ per cent.}$$

The above description of power applies to modern reciprocating machinery, and must be modified for a turbine installation ; in this latter case no means exist for estimating the power developed in the turbine cylinders themselves, and the power is therefore obtained from observations of the distortion of the shafts abaft the turbines, the result

Fig. 283.



so obtained being termed shaft horse power (See Chapter XXVIA). The power given to the propellers is the S.H.P. less the power lost in friction at bearings between propeller and torsionmeter wheels, which may be taken to be from 2 to 3 per cent. at full power and proportionately as the revolutions at intermediate powers.

At the present time when the design of fast-running screws necessary for turbine installation is not understood to the same extent as for the ordinary slow-running screws, it would appear from the few cases in which data are available that the effective horse-power does not greatly exceed 50 per cent. at full power, and the distribution of work in such a case may be approximately taken to be as follows :—

	Per cent.
Loss in friction of shaft bearings	2
Loss at propeller, including augmentation and gain due to wake	48
Balance, or effective horse-power	50

Curves of coal consumption. Economical speed.—It is a usual custom for engineers to tabulate, for their own information, the average horse-power, coal, etc. required when the engines are working at various speeds. These tabulated results only represent a number of isolated facts, but when they are shown in the form of a curve the general law underlying the facts can be more readily ascertained and more valuable information obtained. The construction of curves for all such records is recommended, especially as regards the consumption of coal per day corresponding to various rates of speed of ship or revolutions of engines. A number of runs must be made at various speeds and the coal used carefully measured, the various points ascertained being set off on a diagram and a fair curve drawn through them. This curve, like the thrust curve, if produced to the axis does not intersect it at the zero point.

Such a curve will give by simple measurement the consumption of coal per day for any intermediate speed. One such curve is shown in Fig. 283. It is also easy to ascertain the most economical speed for the engine and the coal consumption at that speed. For if we draw a tangent from O to the curve meeting it at A, it is evident that the ratio of $\frac{\text{coal consumption}}{\text{speed}}$, i.e. the tangent of the angle AOX is least at the point A, and is greater for any other point on the curve, so that A will be the most economical speed of the engine. It will be seen, however, that the range of speed about this point at which the consumption is practically unaltered is considerable, and this being so it will generally be advantageous to select the higher limit of speed.

When the steaming distance of the ship for a given quantity of coal is under consideration, the amount of coal necessary to be expended for purposes other than the main engines must be taken into account. This is done by drawing a line O′X′ at a distance below OX equal to the consumption per day for auxiliary purposes, such as electric lighting, culinary purposes, distilling drinking water, &c. The most economical speed of the ship is therefore obtained by drawing a tangent from O′ to the curve meeting it at B, which gives a greater speed than if the consumption for auxiliary purposes were not taken account of. The reason why the slowest speed is not the most economical will be understood from what has gone before; it lies in the increased proportionate waste by constant friction as the power is diminished, and the proportionately increased loss by radiation from boilers, steam pipes, &c., and the greater proportionate consumption of coal for the constant auxiliary services of the ship.

It will be noticed from the nature of the curve that a reasonable increase or decrease from the economical speed does not materially affect the total consumption of coal, and also that for a certain total consumption there are usually two speeds at which the ship may travel.

In warships, the difference between light and deep load conditions is not so marked as in merchant vessels of large cargo capacity, but it is considerable, and the preparation of curves of coal consumption, horse-power and the corresponding revolutions for various speeds at various draughts, as opportunities occur, is of use to the engineer

in determining the capabilities of the ship, as in Fig. 283a. A rough estimate of the additional fuel due to head winds and seas at a particular draught can be formed by the inspection of the revolution and speed curves.

Amount of consumption for auxiliary purposes.—The consumption of coal on board warships for auxiliary purposes—i.e. for purposes other than propelling the vessel—is very considerable. This service includes the consumption for culinary purposes, warming ship, distilling, electric lighting, working guns, &c., and often amounts during the whole of the commission of a warship to more than that expended in propelling

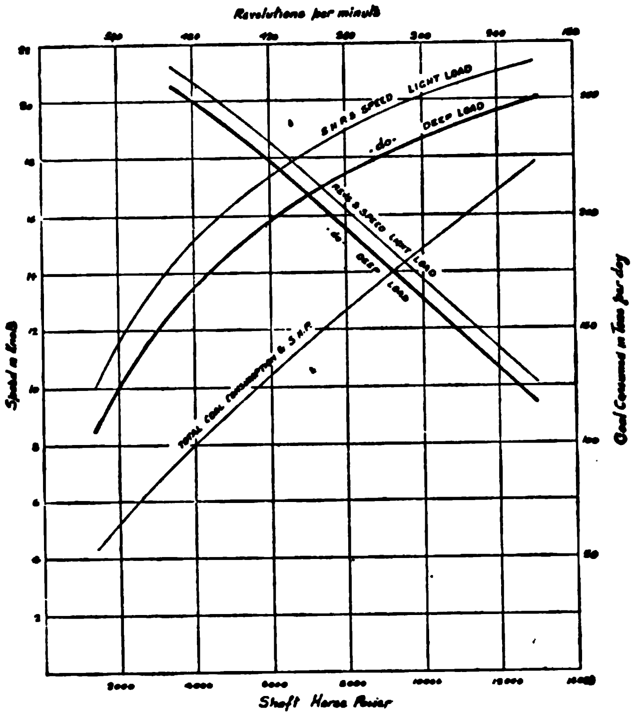


FIG. 283A.

the vessel, owing to the intermittent nature of the steaming generally required. The following amounts are the daily consumptions of coal for auxiliary purposes in modern war vessels: Large 1st class battle ships, 10 to 15 tons; 1st class cruisers, 8 to 15 tons; 2nd class cruisers, 5 to 8 tons; and 3rd class cruisers, $3\frac{1}{2}$ to 6 tons. The actual amount depends on the number and extent of the auxiliary machinery fitted, and tends to increase in the newer ships owing to the larger quantity of fresh water used, extensive installations of lighting and electrically-driven machinery, fitting of refrigerators, &c.

The coal required to run one 200 kilowatt dynamo for battle ships of British Navy for 24 hours at full power has been found to average

8 tons, but even at this rate the cost is much less than if candles or oil were used.

Closed exhaust system.—In the newer ships of British Navy arrangements are made so that the heat of the considerable quantity of steam used for the auxiliary engines is not wasted in the auxiliary condenser, but utilised in the evaporators to make fresh water for the boilers and ship's company. A connection is made between the coils of the evaporators and the auxiliary exhaust pipe for this purpose, the valve on the exhaust pipe at the auxiliary condenser being closed. By this means a pressure of 10 to 15 lbs. can be maintained in the auxiliary exhaust system, the steam being condensed in the evaporator coils, producing fresh water. Piston relief valves are also fitted so that any excess of exhaust steam beyond that required for evaporation passes into the auxiliary condenser when in harbour, or into the L.P. receiver when the main engines are under way. By this means a saving is effected ; the loss due to the back pressure on the auxiliary engines is more than counterbalanced by the fresh water obtained.

CHAPTER XXIV.

PADDLE-WHEELS.

Radial paddle-wheel.—The simplest form of paddle-wheel is generally known as the common or radial paddle-wheel. In this wheel the floats are bolted direct to the arms of the wheel, and consequently the pressure they produce on the water is always perpendicular to the radius, and the only float that produces a direct sternward reaction is the one at the bottom of the wheel, all the others having a vertical component tending to raise or depress the vessel, which is wasted so far as propulsion is concerned.

Width of floats.—The extreme width of the floats should not exceed one-half the width of the vessel, so that the combined width of the two paddle-wheels should not be greater than the width of the ship. In sea-going steamers the width of float generally does not exceed one-third the width of vessel. In still water the greater the width of float the more effective the wheel, as the required area of race can be obtained with less immersion, and the loss from oblique action is thereby reduced. This condition, however, is limited by the practical difficulties involved in supporting the overhanging end of the paddle-shaft. In rough weather extreme width would be objectionable from many causes.

Immersion of wheels.—The depth of immersion of paddle-wheels is practically limited by the draught of water of the vessel, as it is evidently undesirable to allow the lower edge of the propeller to be below the keel. The immersion of the wheels must also depend on their diameter, for if the floats act too obliquely on entering and leaving the water, a large proportion of the power would be wasted in producing vertical reactions. As an extreme case, we may point out that a radial paddle-wheel immersed to its centre would be of no value as a propeller.

In general the greatest immersion of a paddle-wheel should not exceed one-half the radius, or one-fourth the diameter of the wheel. When sea-going steamers were used for long voyages, the immersion at starting was about one-half the radius, and the mean draught for the voyage about one-third the radius of the wheel.

For effective working, the tops of the floats, when in their lowest position, should always be some distance below the surface of the water. In large sea-going paddle steamers the top of the lowest float was usually about 18 to 20 inches below the surface, at mean draught; in smaller vessels from 12 to 15 inches. In river steamers the immersion is generally much less, say from 3 to 6 inches; but these boats always work in smooth water, and their draught is practically constant. In sea-going

steamers the immersion of the floats at their lightest draught should not be less than 6 inches.

Number and pitch of floats.—In radial paddle-wheels the number of floats is generally made equal to the number of feet in the diameter of the wheel, which practically sets them at about 3 feet apart from each other. In some fast ships, to reduce vibration, they have been set closer than this, or from 2 to $2\frac{1}{2}$ feet apart. If the floats be set too closely together the water will not escape with sufficient freedom from between them, whilst if too far apart the vibratory action will be excessive. The number and pitch of floats should be so arranged that there will always be at least three floats immersed at the same time.

Reefing paddle-wheels.—The floats are secured to the radial arms of the paddle-wheels by hook-bolts, in such a manner that if the draught of the vessel be increased, the floats may be readily unshipped and secured in other positions nearer the centre of the wheels. This operation is usually called *reefing the paddle-wheels*, and is equivalent to reducing the effective diameter of the wheel and the immersion of the floats, and thereby diminishing the loss from oblique action. Reefing is desirable when by increased draught it is found that the wheels cannot be driven fast enough to utilise all the steam generated in the boilers. This operation, by decreasing the resistance, enables all the steam generated to be used, and the piston speed increased, with a consequent gain in the power and speed of the ship.

The only points of advantage of the radial over the feathering paddle-wheel are its lightness, simplicity, and cheapness of construction. There are no working parts in it, and defects can be readily made good at little cost. Its propelling efficiency, however, is much less than that of the wheel with feathering floats, and the improvements in design and workmanship have made the latter so practically trustworthy, for the comparatively few services for which paddle-wheels are now required, that the radial paddle-wheel may be regarded as altogether a propeller of the past.

Feathering paddle-wheel.—In order to obviate the disadvantages resulting from the oblique action of the floats of radial paddle-wheels, especially in cases where the draught of the vessel varied considerably, feathering paddle-wheels have been introduced. The general form and arrangement of these propellers are shown in Figs. 284 and 285. The wheel consists of a wrought-iron framework, secured to a strong cast-iron centre or boss, keyed on the end of the paddle-shaft. The floats, instead of being fixed to the arms of the wheel, are carried on joint-pins, and their motion is controlled by the action of an eccentric, through rods and levers, in such a manner as to keep the floats approximately normal to the effective surface during their passage through the water, so that the whole of the thrust will be in a sternward direction. Its efficiency is at least 10 per cent. greater than that of the radial paddle-wheel when both work under suitable conditions, and the economy and efficiency resulting from its use far more than compensate for its increased first cost and expense of maintenance.

It is however more complicated, and requires more care and attention, than the radial wheel. It is very important that the working parts should be sufficiently strong to withstand the shocks to which

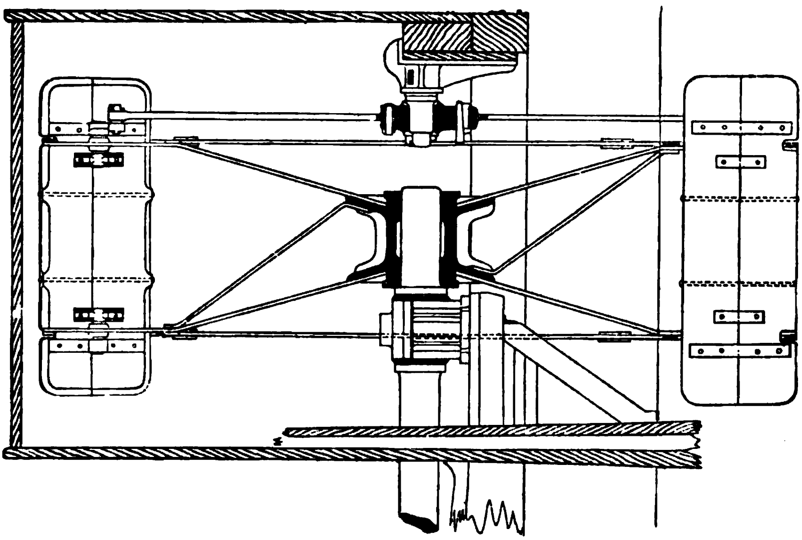


FIG. 285.

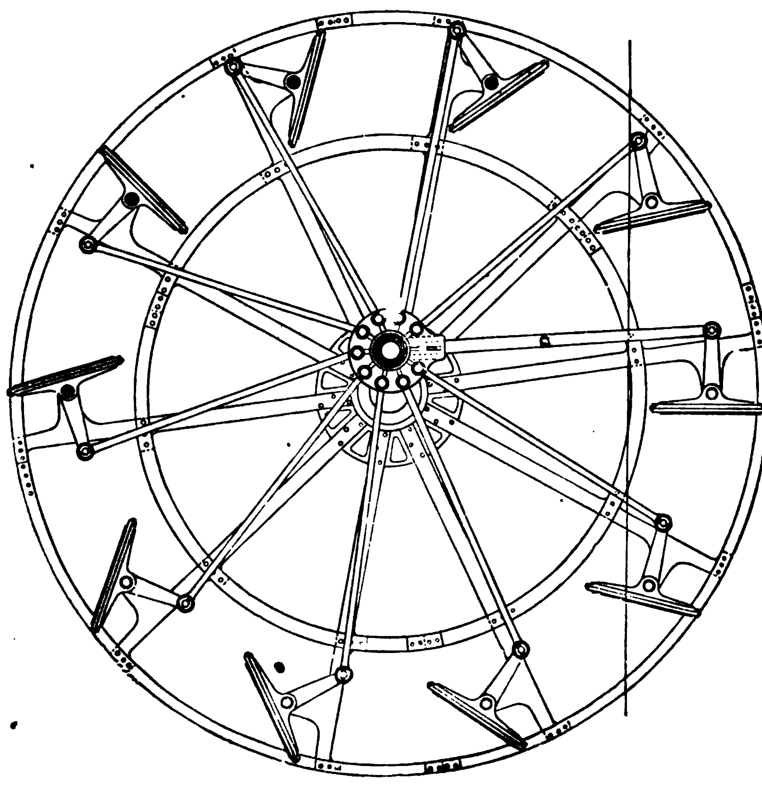


FIG. 284.

they are exposed, without undue straining, for damage to any part of the feathering apparatus is liable to paralyse the action of the entire wheel. These wheels are consequently made much heavier than the radial wheel, and are more difficult to properly support.

This complication and liability to serious injury might possibly have tended to prevent their being extensively used for long sea voyages, in preference to the simpler radial wheels, which, if damaged, could be so much more easily repaired. As, however, the paddle-wheel for ocean navigation has been entirely superseded by the screw-propeller, this point need not be further discussed, and there can be no doubt that for short voyages, river navigation, and towing purposes, for which alone paddle-wheels are now used, feathering floats possess very great advantages, enabling the wheels to be made of less diameter and width, and in consequence of their increased efficiency the indicated horse-power of the engines may be proportionately reduced for a given speed.

Dimensions and pitch of floats.—The floats in feathering paddle-wheels are generally placed about twice as far apart as the floats in the radial wheel; that is, the pitch of the floats is usually about six feet. They are also made deeper, say about twice the depth of the common float, for in this case the area of the race, or stream driven back on either side of the ship, is equal to the width multiplied by the depth of the float instead of the width of float multiplied by the depth of immersion, as is assumed to be the case with the radial paddle-wheel.

Eccentricity of feathering apparatus.—The method of determining the throw and position of the eccentric necessary to produce the proper action of the floats in the water may be easily explained by means of a skeleton diagram. In Fig. 286 let A represent the centre of the paddle-shaft, and K the centre of the eccentric pin or sheave that produces the necessary movement of the paddle-floats, the correct position of which is required to be found. For simplicity, the floats are supposed to be jointed at their centres. In practice this is not exactly the case, the joint being just behind the float, and as close to it as possible. In an actual design, this would render necessary a slight modification in the details of the following method of determining the eccentricity, but the deviation is small, and there will be no difficulty in making the required correction when the principles involved are understood. The circle B C D E F G, drawn with A as centre, through the centres, or joints, of the floats, may be taken to represent the paddle-wheel circle. Let W W represent the water-line, B and D being the points in which it is cut by the paddle-wheel circle.

Consider three floats in the positions shown by B, C, and D, one just entering the water, the second at its lowest point, and the third just leaving the water. In order that the motion of the floats through the water should be correct, moving as nearly as possible edgewise, relatively to the water in the paddle race, the directions of the faces of these floats produced, should meet at the point F, at the top of the paddle-wheel circle. If, therefore, from F, the highest point of the circle, straight lines, F B, F C, and F D are drawn, these will represent the directions of the faces of the paddle floats at these respective points.

From the centres of these three floats, B, C, and D, draw the float-levers, B*b*, C*c*, D*d*. These are usually at right angles to the float, and their lengths are about three-fifths of the depth of float. These values are

arbitrary, and subject to convenience in any particular design; but the angle seldom deviates much from a right angle, and the proportionate length of lever given above is generally suitable. Having thus determined the points, *b*, *c*, and *d*, to which the radius rods from the eccentric have to be jointed, it is only necessary to find by plane geometry the centre, K, of the circle passing through them. K will then be the centre, and A K the throw, of the eccentric necessary to produce the required motion of the floats.

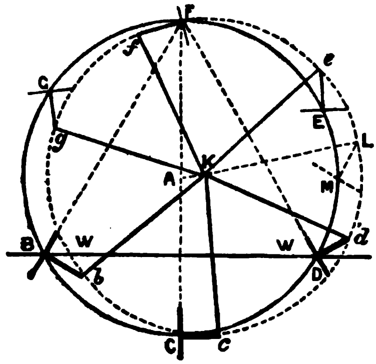


FIG. 286.

The velocity of the propeller race is clearly represented by the circumferential velocity of the circle B D F G, and the effect of the motion produced by the action of the eccentric, thus determined, will be to cause the floats, while *in the water*, to move as nearly as possible edgewise, relatively to the propeller race, and thus prevent loss from oblique motion. By drawing floats in other positions, it will be seen that their action when *out of the water* is far from being free from vertical reactions, but these, operating only on the air, may be neglected.

Paddle-shaft bearings.—The shaft carrying the paddle-wheel is called the paddle-shaft, and is sometimes supported by two bearings, one on the ship's side, and the other on a beam, called the *sponson* or *spring beam*, on the outside of the paddle-box. In this case, the feathering apparatus has to be worked by a large eccentric on the paddle-shaft, to which the radius rods are attached.

Overhung wheels.—The most general arrangement, however, is that shown in Figs. 284 and 285, in which the paddle-wheel is overhung and supported by a single bearing on the ship's side, the outer bearing being dispensed with. In this case the feathering motion is produced by attaching the radius rods to a sheave working on a pin carried by a bracket fixed to the outer side of the paddle-box, in the proper position, eccentric to the wheel, to produce the required movements of the floats.

Driving and radius rods.—In the feathering apparatus, one of the guide or radius rods, called the driving rod, is rigidly fixed to the eccentric, to make it rotate about the axis K. The remainder of the rods are simply jointed to the eccentric, as well as to the float-levers, with pins. In Fig. 284 the driving rod is marked D. All the joints in the feathering apparatus should be bushed either with gunmetal, white metal, or lignum-vitæ.

Details of paddle bearings.—The outer bearings of paddle-wheels, when they are so fitted, cannot be examined when the engines are at work. Guide-boards or troughs are therefore fitted on the side of the paddle-box, so that the water carried up by the wheel is caused to constantly run on these bearings to prevent their overheating. This splashing and churning action of the wheel on the water is also often

the circumference. In Fig. 288, let A B represent the pitch and B C the circumference of a screw, to any given scale. Then the angle A C B will represent the angle of the screw at the end of the blade. The pitch being assumed the same throughout the blade, the angle at any other part of the blade may be found by determining the circumference B D at the given part, and joining A D; the angle A D B being the required angle.

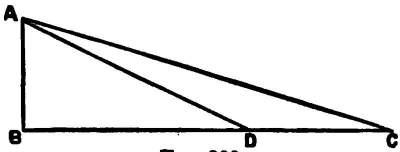


FIG. 288.

Form of blade.—Screw-blades are made of a great variety of forms. The effect of form of blade has not yet been fully ascertained, but the shape suitable for one ship does not always prove equally efficient for a different ship. As a rule it would appear that form has little peculiar value as regards propulsive efficiency, though it may have some influence on the amount of vibration produced; the main points to be considered are, the pitch and surface of the blades in relation to diameter.

The work absorbed in friction and the disturbing effect on the stream-line motions must necessarily have some effect in determining the most suitable shape of the blade, but this has not yet been reduced to exact calculation.

In consequence of the increased angle of the screw-blade as it approaches the axis, the inner part of the blade, if the boss be small, has very little propulsive efficiency, and only absorbs power in churning the water. This was very noticeable in the earlier forms of screw-propellers, in which the boss was only about twice the diameter of the shaft, so that the inner portions of the blades were nearly in a fore-and-aft direction. In these screws also, the length of the screw, or, in other words, the length of the projection of the blades on a fore-and-aft plane, was constant throughout the blades, so that the side view of the screw was rectangular. The ends of the blades were, therefore, very broad, and absorbed much power in surface friction, owing to their great velocity.

An endless variety of different patents have been originated on the question of the form and arrangement of screw-propeller blades, many of them peculiar and complicated, but they have all failed on trial to give the results anticipated.

Hirsch screw.—A variety of screw much favoured at one time was the Hirsch screw, a four-bladed example of which is shown in Fig. 289. In this propeller the blades are curved forward, the axis of the blade being approximately a spiral curve, instead of a straight line as usual, the pitch also increases towards the circumference, and there are other minor peculiarities. The method of obtaining the curve of the centre line of the blade is shown by the dotted construction.

It was supposed that this curved form of blade tended to resist the centrifugal motion of the particles of water acted on by the propeller, and that vibration was diminished by the action of the blade being gradual.

Griffiths' screw.—Mr. Robert Griffiths was probably the most successful of the early designers as regards propeller proportions. He

substituted for the central inefficient portion of the screw a large spherical boss, one-third the diameter of the propeller, which would revolve without agitating the water. This principle is now adopted for most screw-propellers, though the bosses are not usually so large as in the earlier Griffiths' screws. The general diameter of the boss in screw-propellers as now fitted is about one-fifth to one-quarter the diameter of the propeller. In the Griffiths' screw also, the outer ends of the blades, which revolve at the highest velocity, were considerably narrowed, to reduce loss from friction. The widest part of the blade was about four-tenths of the radius from the centre, the blade being somewhat pear-shaped. The tips of the blades were bent forward to the extent of about one-twenty-fourth of the diameter of the propeller.

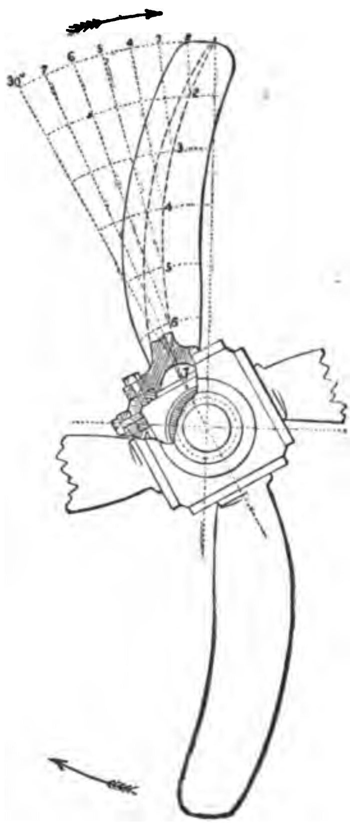


FIG. 289.

The two-bladed propeller shown in Fig. 290 is an example of a somewhat modified form of Griffiths' screw fitted to a vessel with a lifting screw.

Propeller arrangements for old masted ships.—Before describing the modern arrangement of propellers and shafting, some space will be devoted to that of the early single-screw vessels, which were supplied also with sail power. Such vessels were intended at times to proceed under sail alone, and disconnecting couplings were fitted to enable the propeller to revolve freely without moving the engines. This was usually effected by fitting one of the couplings with bolts, that could be readily withdrawn by means of screws. A friction strap was fitted to hold the propeller during the time the bolts were being withdrawn or replaced, and a thrust-bearing abaft the disconnecting coupling prevented the shaft being drawn back by the propeller when revolving while disconnected.

The resistance offered when under sail by the propeller was still great, and to obviate this one of the following plans was adopted :—

1. To lift the screw entirely out of the water.

2. To arrange the *blades* of the screw so that they could be 'feathered,' that is, turned round on the boss, so as to be approximately in the fore-and-aft direction.

3. The fitting of 'Mangin' screws with narrow blades behind each

other, to reduce their transverse width and enable them to be more completely masked by the stern post.

Lifting screws.—The earlier single-screw ships in the Royal Navy were fitted on the first plan. On the end of the screw-shaft a gun-metal coupling called the *cheese-coupling* was keyed, having a rectangular slot. The boss of the screw was cast with journals on its forward and after sides. The foremost journal had a T-head fitting into the slot in the cheese-coupling, by which the rotation of the shafting was transmitted to the propeller when in place. These journals were carried by bearings, fitted with strips of lignum-vitæ, in the ends of a frame called the 'banjo-frame.' Arrangements were made so that when the slot was turned in the vertical direction, the banjo-frame carrying the propeller could be lifted clear of the water.

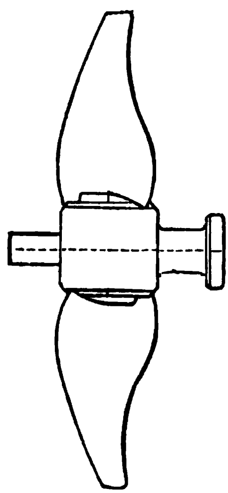


Fig. 290.

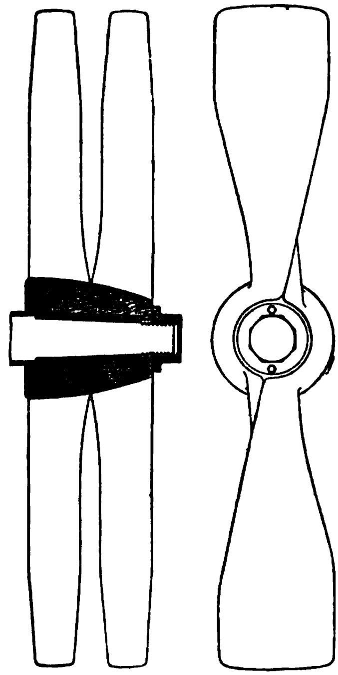


Fig. 293.

There were many serious objections to the application of lifting screws, also many practical difficulties attending their management and efficient maintenance, so that attention was soon turned to the design of arrangements enabling sails alone to be used without undue increase of resistance from the propeller when fixed vertically. The second and third of the above plans were therefore introduced.

Feathering screw.—With this the blades could be turned in an almost fore-and-aft direction, so that they produced much less retarding effect on the vessel. The blades were turned round by a lever, worked by a small rod carried through the centre of the stern-shaft, which was hollow, to the forward end of the shaft, where means were provided for moving the rod to and fro.

Mangin screw-propeller.—The Mangin screw-propeller is shown in Fig. 293. It may be regarded as a two-bladed propeller in which each blade is cut in halves, with one half set immediately behind the other on the shaft, so that the width it occupies is only one-half that of an ordinary two-bladed screw, and its retarding effect on the ship would be correspondingly reduced. When the aggregate area of the four blades is equal to that of the two blades of an ordinary screw, there is little difference in the efficiency.

Modern screw-propellers.—The arrangements for providing for the application of sail power in steam vessels are now practically obsolete. Twin, triple, or quadruple screws are now universal in the large vessels of the Royal Navy, and as sail power is thus not required to provide for possible breakdown of the machinery, the masts and sails, which are inconvenient and interfere with the fighting efficiency of a warship, are not now fitted. The following paragraphs describe the usual screw-propellers now fitted to various classes of vessel.

Number and pitch of blades.—The usual number of blades in screw-propellers are two, three, and four. Three- and four-bladed propellers appear equally efficient and suitable for large vessels. The blades are, except in small screws and fast-running propellers for turbine machinery, cast separately from the boss, and are secured to it by bolts, generally forged of naval brass, manganese bronze, &c. Three- and four-bladed screws are now the most general. Two-bladed screws used to be fitted in cases where it was intended to lift the propeller out of the water when the ship was under sail.

The four-bladed screw-propeller with a single-screw ship has the effect of reducing *vibration* compared with a two-bladed screw, being more continuous in its action. There appears to be no advantage from any point of view in increasing the number of blades beyond four.

The pitch of the blades is generally uniform, but sometimes the pitch of the leading half of the blade, or the part that first acts on the water, is made less than that of the following half, to make the action gradual and decrease shock.

Material of propellers.—In the mercantile marine, screw-propellers are often made of cast-iron, for cheapness, the blades being cast on the boss so that the whole propeller forms a single casting. In the Royal Navy, and often in the mercantile marine, the screws are generally made of gunmetal, manganese bronze, phosphor bronze, &c. A cast-iron boss with manganese bronze blades is a common fitting in the mercantile marine, while in vessels such as torpedo-boat destroyers a steel boss with blades made of some special alloy of bronze has been used.

Screw-propellers can be made much thinner and lighter of gunmetal than of cast-iron, and still lighter if made of manganese bronze or some similar alloy. The surfaces of the blades should be made as smooth as possible to reduce friction, and the use of brass or bronze enables the surface to be readily smoothed off and polished. By this means the edgewise resistance is reduced, and by making the blades separate from the boss the whole screw is not destroyed in the event of damage to a single blade. Spare propeller-blades, or propellers, are always provided.

In some ships cast-steel is used for screw-propellers to thin the blades, but their backs have been found to suffer considerably from corrosion. This has been remedied in some cases by fitting a brass sleeve on the blade at this part.

Alteration of pitch.—The holes in the blade flanges of large propellers are made elongated, to allow the pitch of the screw to be adjusted slightly if necessary, by turning the blade round on the boss and fixing it in a different position. The extent of the variation of pitch allowed is from 2 to 3 feet. Filling pieces of brass or lignum-vitæ are supplied, for the spaces on either side of the bolts.

A slight alteration of the pitch is often found by the steam trials of vessels to be necessary to secure the best results. It is also desirable in some cases, especially in old vessels, when the working pressure of steam in the boilers is reduced as they become worn, for the *weight* of steam that can be produced by the boilers is practically constant for moderate alterations of pressure, and consequently, at the reduced pressures, the *volume* of the steam generated will be greater than at the original pressure. If it be desired to utilise all the steam at the lower pressures, it is generally necessary to drive the engines faster, and to do this the pitch of the screw requires to be reduced. With the much greater life of modern boilers, and having in view the margin of power generally existing when new, this operation is now much less common.

Shape of blades.—The acting surface or '*front*' of the blade, that is, its after face, preserves the exact geometrical form; the form of the forward face or '*back*' of the blade is modified by the thickness necessary for strength, and is not a true screw-surface. The transverse sections of the blades approximate in shape to semi-ellipses, shallow at the ends and becoming fuller towards the root. The sections at various radii of a modern screw-propeller, and the angle of the screw at that section, are shown in Fig. 297.

As regards the form of blade, although the numbers of ideas on this point are innumerable, yet gain does not appear to be effected in practice by departing much from the elliptical form with the extremities of the transverse axis at the centre and tip respectively, as shown in Fig. 297. This is now the general form of the expanded surfaces of the blades of H.M. ships, and it has given more satisfaction on the whole than other shapes.

Propeller diameter and pitch.—Many rules have been proposed for determining the most suitable diameter, area, &c., of screw-propellers, but they fail to be of much service unless the conditions are very similar, so that most propellers are now designed from experience obtained in previous similar vessels, and the results of trials made with differently proportioned screws. In the Royal Navy experiments with model screws are made in the large tank at Haslar, and most of the screws for large naval vessels are now proportioned from trial data obtained from this source.

These experiments are made on screws of elliptical form with the extremities of the axes at the centre and circumference of the screw respectively, the breadth of the ellipse being one-fifth the diameter. It appears that there is a wide range of 'pitch ratio,' i.e. the ratio of pitch to diameter, which can be adopted without material change of

efficiency, provided in all cases that the amount of blade area allowed is properly proportioned to the pitch ratio adopted. As the diameter is reduced, so the pitch should increase for the standard elliptical blade, and the blade area will also be reduced.

The power and revolutions of the engines being first determined, and the speed estimated, the results of the experiments may be indicated in the form of a curve, a sample of which, for a large battleship, is shown in Fig. 294. This curve gives the corresponding pitches and diameters for either three-bladed or four-bladed propellers of the standard shape, the pitch being practically the same for either three- or four-bladed screws. Any diameter inside the limits shown can be adopted, provided the corresponding pitch given by a vertical ordinate is also used. For example, the dotted vertical line A B C shows that for a pitch of

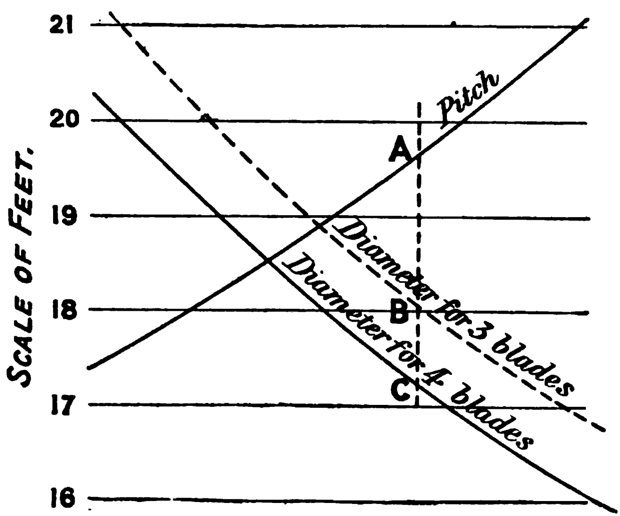


FIG. 294.

19·6 ft. the corresponding diameter for a four-bladed screw is 17 ft. 3 in., and for a three-bladed screw about 18 ft. It will be seen that the variation of pitch and diameter permissible is considerable. Generally a point near the middle of the curves is selected. Knowing the shape of the standard blade and the size of the boss, the blade area can then be calculated.

With torpedo boats and destroyers the system of actually trying various propellers has been found essential, and it is very remarkable how appreciable an effect even apparently small changes in propellers have on the speeds of the boats.

With turbine machinery higher speeds of revolution are necessary to obtain a suitable economical steam consumption which increases considerably as the revolutions decrease. On the other hand, the efficiency of the propeller falls off as the revolutions are increased, and in order to obtain sufficient blade area for the necessary thrust with normal proportions of propeller it becomes necessary to increase the

number of propellers and shafts. The general proportions of propeller and turbine which give the best overall efficiency of propellers and turbines has still to be determined by experience. Efforts are being made, however, to increase the combined efficiency by the adoption of turbines of the impulse type which can develope their full power at revolutions suitable for the use of twin screws, in lieu of the triple and quadruple screws at present required by turbines of the reaction type. With a similar object, and also to allow the turbines to run at a high number of revolutions when the vessel is steaming at low speeds, practical experiments on a large scale are being made with mechanical gear wheels connecting high speed turbines with low speed shafts. Systems of high speed turbo-electric-generators driving low speed electric motors on the propeller shaft, and of hydraulic power transmission, have been considered also, but experience alone will show which of the various methods will eventually be the most successful, numerous advantages and disadvantages being claimed for each system.

Expanded surface of blades.—This is the surface obtained by assuming the actual width of the blade at various distances from the centre to be laid off on a plane surface; this area may be imagined to be obtained by flattening or untwisting the propeller-blade, and is shown at A in Fig. 297. The area provided is as follows in the recent vessels for His Majesty's Navy :—For four-bladed propellers of battleships, ·012 to ·014 square foot per I.H.P. ; for three-bladed propellers of quick running cruisers, ·007 to ·009 square foot per I.H.P., while for the quicker running torpedo-boat destroyers it varies between ·005 to ·007 square foot per I.H.P.

In the mercantile marine the areas are larger. They vary considerably, depending on the speed. From general considerations the thrust varies inversely as the speed, i.e. if two vessels are designed for the same horse-power, and the speed of one is double that of the other, the propeller thrust in the faster vessel is about half that in the slower ship. The developed area depends on the thrust to be exerted, and pressures of about 10 to 11 pounds per sq. inch on the projected area have given satisfactory results in practice. For ocean single-screw passenger steamers, however, ·02 sq. ft. per I.H.P. is a common value, while ocean cargo steamers have considerably more area, ·06 sq. ft. per I.H.P. being not uncommon.

Strength of propeller-blades.—The propeller-blade should be strong enough to withstand its share of the twisting moment on the shaft, but it should be weaker than the shaft with reference to the straining action of a blow from a hard substance on the tip of the blade. The moment of resistance of the blade in going from the boss to the tip should also diminish faster than the moment of such a blow, so that if fracture resulted it should take place as near to the tip of the blade as possible, so that the broken blade may be still useful for propulsion.

In Fig. 295 let A represent the axis of the shaft, the circle G D H the boss of the propeller, and A B a radius drawn from centre of shaft to tip of blade. Let A C, perpendicular to A B, represent the calculated thickness of blade near the axis, and join C B. Then the triangle B A C will represent the section through the centre of a blade that fulfils the condition of gradually diminishing in strength from the centre to the

tip, since the strength of any section varies as the square of the depth. In practice the end of the blade cannot be reduced to a point as shown; so from B, a distance B E must be set off equal to the minimum practicable thickness, and a line E F drawn parallel to the face A B. The point F will then be the point at which it will give way if fractured by a blow on the tip.

The determination of the thickness of the blade is largely a matter of experience. In modern fast-running propellers the centrifugal effects are sometimes pronounced, and the general tendency is to keep the blade erect, as in Fig. 297. It is advantageous from this point of view to allow the blade a certain amount of lag, as in Fig. 300, in which case the centrifugal action tends to assist the blade in withstanding the thrust and turning forces.

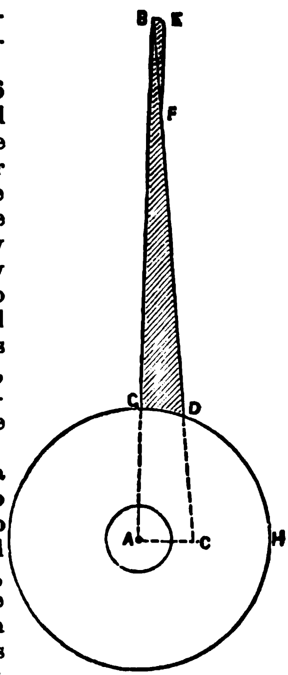


FIG. 295.

Details of modern propellers.—Figs. 296 to 298 show the usual form of four-bladed screw-propeller now fitted in the Navy. The blades are bolted to a boss, about a quarter the diameter of the propeller, secured on the end of the screw-shaft. The hole through the boss, and the shaft, is tapered, and the screw is driven by the action of a longitudinal key or feather let into the shaft, and fitting into a suitable key-way cut in the boss. The end of the shaft is screwed, and the propeller boss is kept in its place by means of a cap-nut, secured by a keep-plate, which prevents corrosive action of the water on the end of the shaft.

The nut and end of the propeller have a conical tail-piece secured over them, to reduce the loss by eddying motion and to prevent fouling of ropes, &c. The gunmetal liner on the shaft is recessed into the boss, and a small stuffing-box fitted to prevent the access of water to the shaft at its junction with the propeller. The holes in the flanges of the blades are elongated, as previously described, to enable the pitch of the screw to be adjusted, and brass or lignum-vitæ stops are fitted between the bolts and the edges of the holes to prevent the blade shifting.

The flanges of the blades are recessed into the boss, and the heads of the bolts securing the blades are recessed into the flange, these latter recesses being covered with plates. Keep-plates are fitted between the bolt-heads to prevent any slacking back. By this means the spherical form of the boss is preserved, and resistance in the water reduced.

The whole of the propeller, except the bolts, is usually of gunmetal, the bolts being made of some forged metal, such as naval brass. The blades are often of manganese bronze. The three-bladed screws are of similar construction.

Fig. 299 shows a three-bladed propeller as fitted in the mercantile

marine. Its construction is somewhat similar to the preceding. The boss is, in this case, of cast-iron, the blades are of manganese bronze,

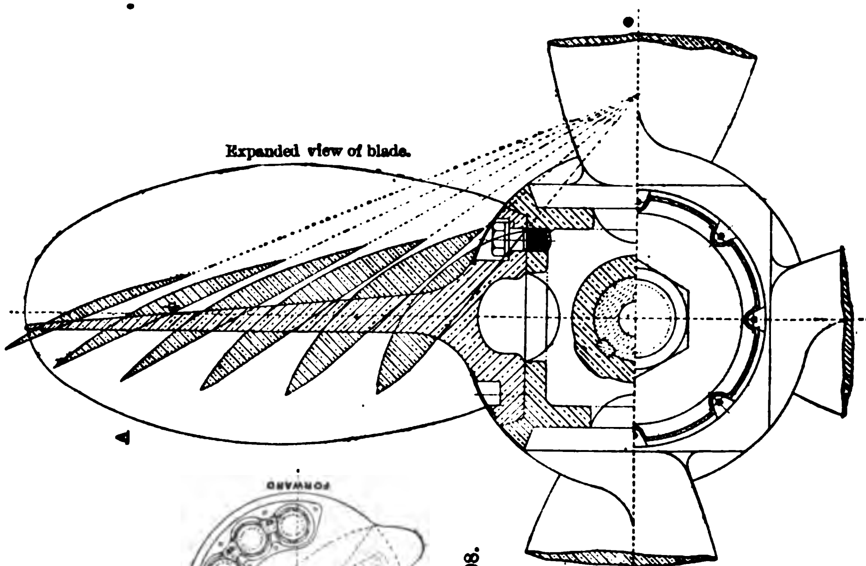


Fig. 297.

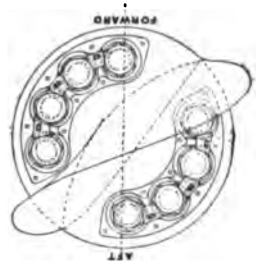


Fig. 298.

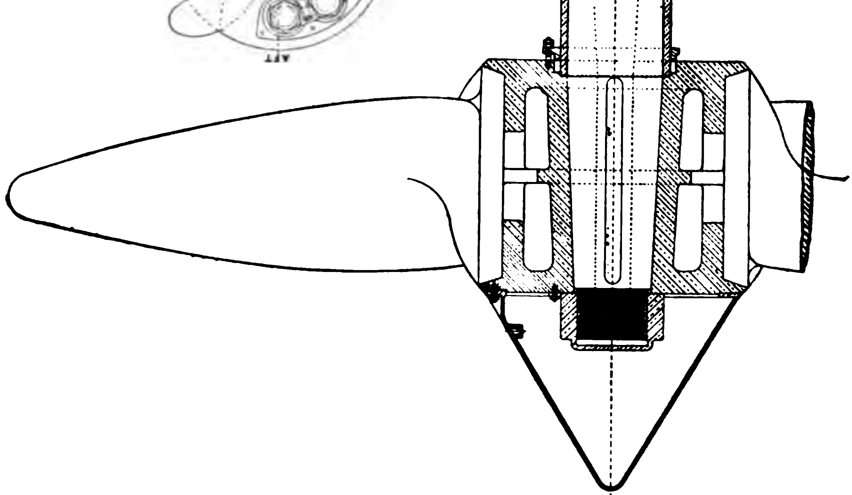


Fig. 296.

and provision is made for slightly altering the pitch by elongating the bolt-holes.

Fig. 300 shows a propeller of entirely different construction, and represents that of a torpedo-boat destroyer, making about 400 revolu-

tions per minute. These screws are generally three-bladed, and have a certain amount of slope astern as we proceed from boss to tip. The blades in the example shown are keyed to the boss, but they are often

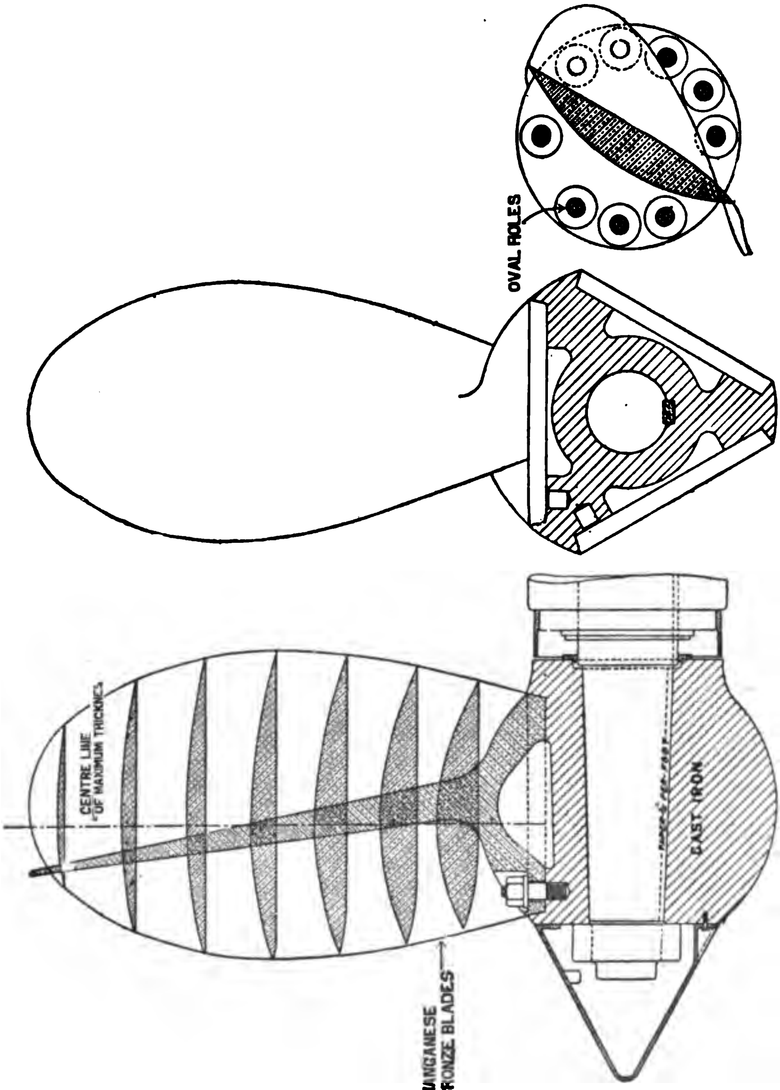


FIG. 299.

cast in one with it. Fig. 300A depicts a propeller cast solid in bronze for use with a turbine-driven destroyer.

Stern-tube.—When the shafting passes through the hull of the ship and enters the water, special means are provided to insure the watertightness of the hull, and to obtain sufficient support for

the shafting. This consists of the '*stern-tube*,' which, in ships of the Royal Navy, is generally made of gunmetal, while in merchant steamers it is generally made of cast-iron. In wood and composite ships the stern-tube is fitted in a wrought-iron or steel casing built into the stern of the ship, and bored out to receive it. This additional casing is also fitted in the Royal Navy to most steel ships with twin screws. Figs. 301 to 303 show the stern-tube.

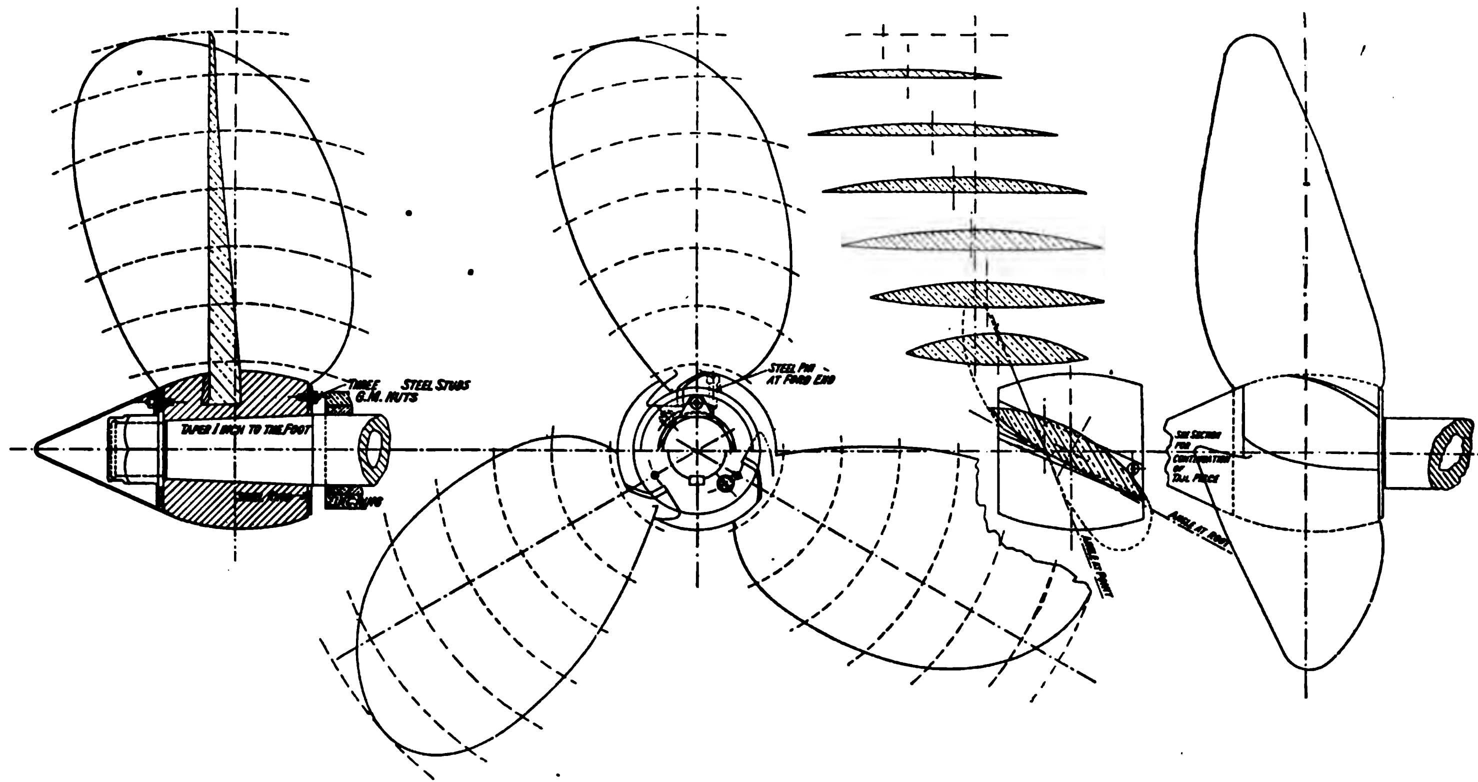
The outer casing or tube is called the shipbuilder's stern-tube, and consists of a tube of steel plate built into the framework of the vessel. At the ends, where the engineer's stern-tube bears on it, steel stiffening bushes are fitted. These are bored out to fit the bearings on the engineer's tube. The bushes are attached each to two frames of the vessel, specially stiffened to carry and distribute the weight of the shafting. With twin screws, where the shafting leaves the ship at the side, the frames and plating are bossed out to surround the stern-tube for some distance aft. The stern-tube is generally placed in its bearing from inside the ship, a flange being provided on the forward end for the attachment to the ship.

The bearings, or rubbing parts, of the stern-tube are fitted with strips of lignum-vitæ, between which the water can pass freely to lubricate the shaft, the lower strips being generally arranged so that the shaft rests on the end grain of the wood. Originally brass bearings were used, but it was found that they wore away very rapidly, as with metal on metal under water, the pressure that can be safely carried is much less than with lignum-vitæ. The lignum-vitæ at the after end of the tube is now often carried in a separate bush made in halves, which can be withdrawn for examination at any time without removing the shaft, the halves being made slightly taper or stepped, to assist withdrawal.

To prevent water passing into the ship, a stuffing-box is fitted where the shaft leaves the stern-tube at the inner end, with a gland known as the 'stern-gland' which, when the engines are at work, is slackened to allow a little water to run through and keep the rubbing parts cool. The cock with passage, as shown, is kept partly open when under way to allow water to circulate and to detect any increase of temperature. Gear is fitted to the stern gland to insure all the nuts being screwed up and slacked back equally. The six nuts and pinions are marked A in the figure, while the keeps for securing the circular rack are marked B, and are three in number. Fig. 304 shows the stern-tube of a single-screw mercantile vessel, which will be readily understood from the preceding description. More recently the shipbuilders' stern-tube has been made of cast steel with additional thicknesses at each end, in lieu of the engineers' tube, to facilitate boring operations.

In high speed vessels it is sometimes found that the speed interferes with the supply of cooling sea water through the tube, and means are frequently fitted to enable the sea water to be pumped outwards through the restricted passages between the lignum-vitæ strips by pumps in the engine room. To further assist in the lubrication of these bearings arrangements are sometimes made which permit of grease being forced into the bearings by means of hand pumps.

Stern shafting.—The length of propeller shafting, which passes



THREE STEEL STUDS
G.M. NUTS
TAPER 1 INCH TO THE FOOT
STEEL PIN
AT FORD END
SEE SECTION
FOR
CONTINUATION
OF
TAIL PIECE
ANGLE AT ROOT
ANGLE AT POINT

Fig. 300.

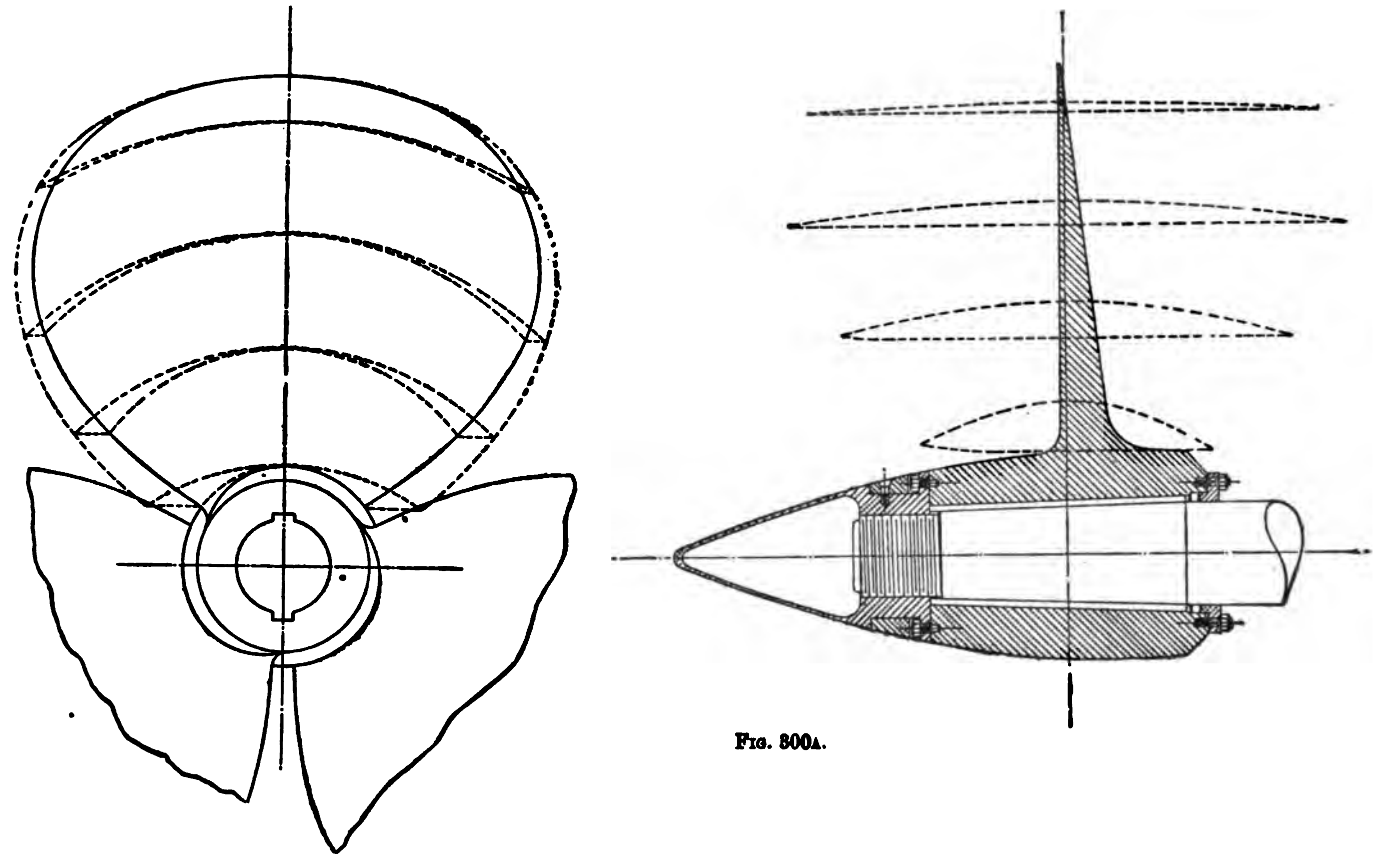


Fig. 300A.

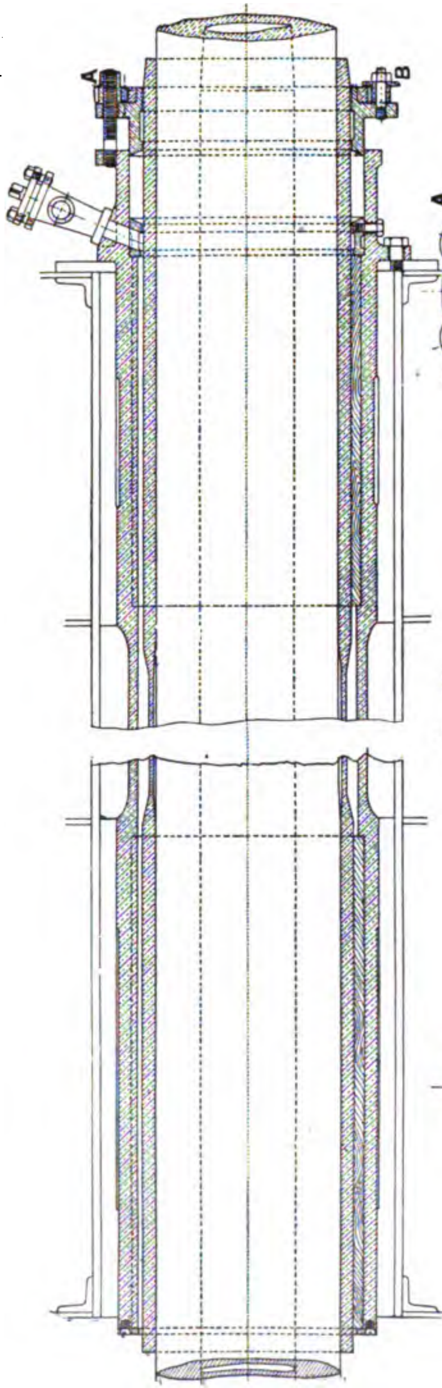


Fig. 301.

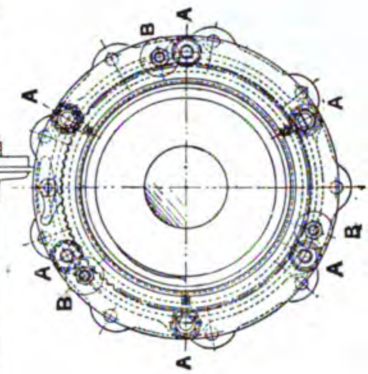


Fig. 303.

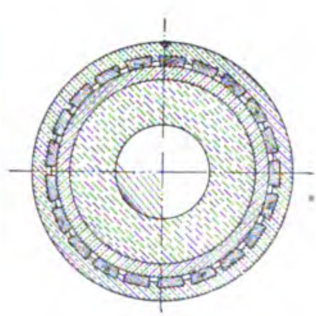


Fig. 302.

through the stern-tube, is covered with a gunmetal sleeve or casing. This is generally cast separate, and after being turned and bored is forced on by hydraulic power, after being slightly warmed. Sometimes this casing is cast around the shaft, which in this case is usually grooved, to prevent the sleeve turning, the shaft itself being made hot before the metal is cast around it, but the greatest care is necessary to prevent the shaft bending and to insure a sound casting. The former method is the better.

The casing inside the stern-tube is in one piece unless it is too long to be so cast, in which case it is fitted in two lengths tightly filleted into each other, and usually brazed at the junctions, for if there should be any leakage, the shaft will decay at the joints from galvanic action. Two methods of forming this joint are shown in Figs. 305 and 306. Studs should be screwed through the casing into the shaft to prevent any change in its position by the working of the engines. The parts of the casing at the ends of the shaft which work in the lignum-vitæ bearings should be thick and fit solidly on the shaft. The intermediate portion, which simply protects the shaft from the action of the water, is generally made much lighter. The space between the liner is subsequently tested with oil and filled with a red lead paint mixture as a further precaution against corrosion. In the mercantile marine, the intermediate portion of the sleeve is frequently dispensed with, the shaft being cased with gunmetal only at the bearings, and the centre part either left bare or lapped with wire and painted. This construction is shown in Fig. 304.

Arrangements for copper sheathed vessels.—If the vessel be sheathed with wood and coppered, the copper exercises a powerful galvanic action on all exposed steel surfaces under water, so that in twin-screw ships it is necessary to cover the whole of the shafting under water with gunmetal sheathing similar to that described for the shafting in the stern-tube. In this case the casing is always in lengths, tightly filleted into one another, and in addition well brazed at the junctions. The space between the shaft and the casing is usually filled with some protective material. Any couplings in the shafting under water will also require to be fitted with a gun-metal casing, but in modern high speed vessels for H.M. Service external couplings are avoided wherever practicable. Sketches of the gunmetal casing over the coupling under water are given in Figs. 307 and 308. Whether the ship be sheathed with copper or not, the part of the shafting which passes through the stern bracket in twin-screw ships is always cased with gunmetal, which works on lignum-vitæ placed in a bush fitted to carry it, in the stern bracket.

Thrust arrangements.—In the older single-screw ships, the propeller being situated at a very strong part of the hull, arrangements were made for part of the forward thrust to be taken directly on the sternpost; a ring or disc, lined with lignum-vitæ, was fitted on the after-face of the stern-post for the forward end of the propeller boss to press against and drive the ship forward (see Fig. 290). The whole of the astern thrust was taken inboard by an ordinary thrust bearing on the shaft such as is described in Chapter XXI. In addition, when lifting screws were fitted, as there was no rigid connection between the propeller and shafting, an additional disc fitted with lignum-

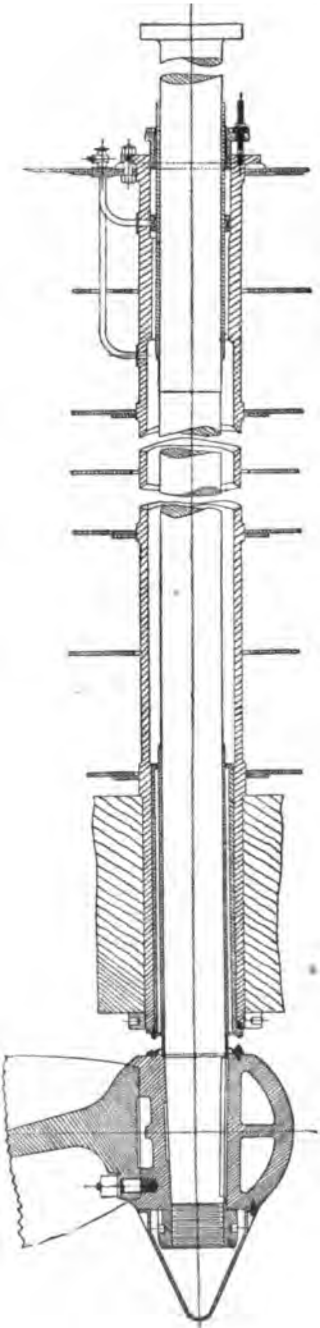


FIG. 304.

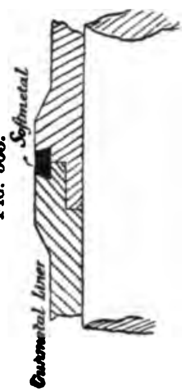


FIG. 305.

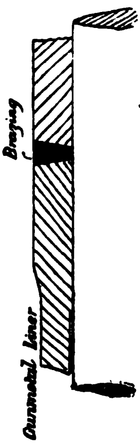


FIG. 306.

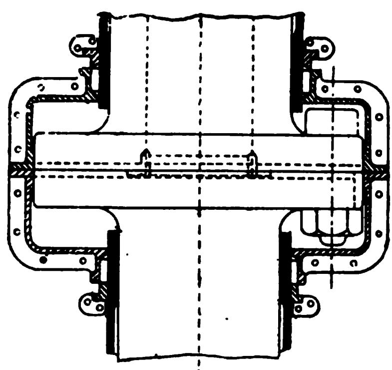


FIG. 307.

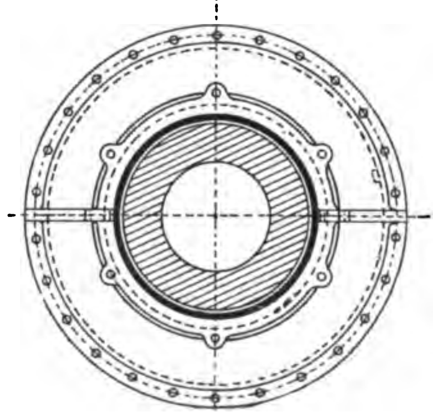


FIG. 308.

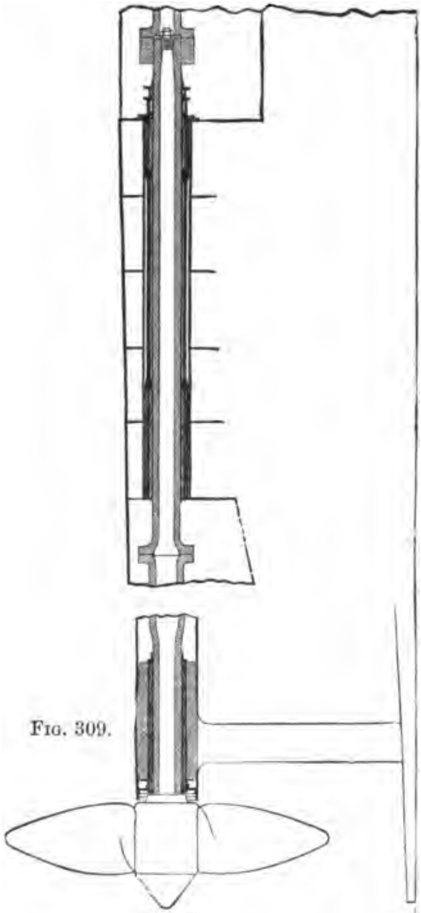


FIG. 309.

vitæ was required, at the after end of the propeller on the rudder-post, to take the backward thrust, the rudder-post being made strong enough for this purpose.

In twin-screw vessels the propellers cannot press directly on the stern, and the shaft brackets are incapable of taking the thrust, so that in such cases both ahead and astern thrusts must be wholly taken inboard, by means of a thrust-block on the shafting.

This system of taking the whole of the thrust both ahead and astern by means of a thrust-block fitted inside the vessel is a satisfactory method, the thrust-blocks being very efficient. They can be always under observation, and can be easily adjusted and kept in proper order. Besides being adopted for twin-screw ships, its convenience has led to its adoption in single-screw ships, so that the single-screw mercantile marine vessels are also generally without any outboard thrust arrangements, and depend on the inboard thrust-block to take the whole thrust. In Fig. 304, showing the stern tube and propeller of a modern single-screw mercantile vessel, there is no outer thrust bearing. In the case of certain turbines the thrust can be arranged to be practically balanced by the steam pressure on the turbine

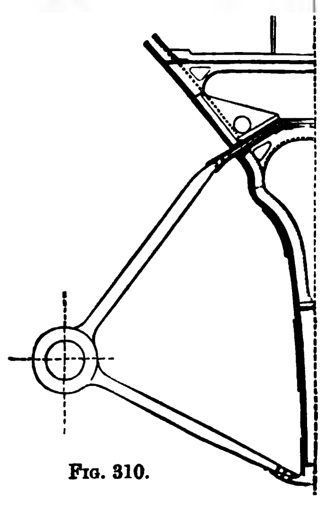


FIG. 310.

rotor, whether going ahead or astern, and the load is then taken off the thrust collars which are retained for purposes of adjustment and safety, the loss of power involved by the rubbing of the surfaces being thereby avoided.

Shaft Brackets.—In twin-screw ships, the propellers on either side of the ship usually work outwards when driving the ship ahead; but in many recent war-vessels, as the result of experiments made in the Admiralty experimental tank at Haslar, they are made to turn inwards. These experiments, made on models, indicated a slight increase in efficiency, so that, as it is more convenient from the engineer's point of view for the screws to turn inwards, the starting platforms being then both in the front of the engines at the middle line, the plan has been adopted in many ships for the Royal Navy. Experience with engines so arranged has, however, caused some adverse criticism as regards the power of the screws to turn the ship when working one ahead and one astern, which appears to be reduced. For turbine machinery in H.M. Service the screws are outward turning. In the Mercantile Marine with turbine lines of high speed the practice has sometimes been to have the wing and inner shafts inward and outward turning respectively.

The after parts of the propeller shafting pass outside the ship and work in bearings close to the propellers, carried by brackets secured rigidly to the hull. In the older high-speed ships, in which the after run is very fine, the length of the shafting outside the ship is so great that intermediate bearings, between the stern-tube and the after-bracket, have been fitted, but the resistance of these additional bracket-bearings is great, and they have not often been fitted to modern vessels. In modern ships, hollow steel shafts, of enlarged diameter between the bearings to give increased stiffness, are fitted, to dispense with the intermediate bearings and their resistance.

Figs. 309 and 310 show the stern and propeller fittings of modern high-speed twin-screw warships. The bush carrying the lignum-vitæ in the after bracket is in these sketches withdrawn by removing the propeller and drawing the bush aft, but as this involves considerable work, these bushes in the more modern vessels are made in halves, and arranged so that it can be withdrawn forward so that the propeller need not be disturbed.

CHAPTER XXVI.

THE INDICATOR AND INDICATOR DIAGRAMS.

We will now proceed to describe the apparatus which enables the engineer to ascertain many facts of the greatest importance as to the action of the steam inside the cylinder. This instrument is called the steam-engine indicator, or, shortly, the 'indicator.' Unfortunately, even when the indicator has told us all it can, regarding the interior economy of the steam-engine, there remains much respecting which our knowledge is very imperfect.

The steam-engine indicator is an instrument which shows the pressure of steam in the cylinder at each point of the stroke of the piston. This pressure varies considerably, and is shown for both the outward and inward strokes, which enables the effective pressure at any point of the stroke of the piston to be ascertained, and the mean effective pressure on the piston during the stroke to be calculated.

General features of indicators.—The general features of the instrument are as follows:—A pencil is attached by means of a system of levers to the piston of a small cylinder of known area, open to the atmosphere at the top, and connected by means of stopcocks and pipes to either end of the engine cylinder as required. When the stopcock is open, so as to place the bottom of the indicator cylinder in connection with one end of the engine cylinder, the indicator piston, carrying the pencil, is moved up and down by the varying pressures of the steam, the motion of the indicator piston being resisted by the action of a spiral spring of known elastic force. A sheet of paper is fixed on a barrel, which is caused to revolve backward and forward in a manner coincident with the motion of the engine piston, and on this moving paper the pencil traces a curve or diagram, from which, at any given part of the stroke of the engine, the corresponding pressures of steam in the engine cylinder may be measured. From the mean effective pressure ascertained from this diagram the I.H.P. of the engines is ascertained. This determination of the horse-power is the principal use of the indicator. By means of the indicator diagram, however, many other particulars relative to the action of the steam in the cylinders, and the adjustment and condition of the slide-valves and pistons, may be ascertained; and many improvements that have been made in the performance and efficiency of steam-engines have been largely assisted by the application of this instrument.

The following important particulars may be seen by inspection of the diagrams:—

(1) Whether the admission of steam is early or late, the amount that the initial pressure in the cylinder is below the boiler or receiver

pressure, and whether the pressure is well maintained up to the point of cut-off or not.

(2) The part of the stroke of the piston at which the admission of steam to the cylinder is cut off, and whether the cut-off is sharp or gradual.

(3) At what point and pressure the steam is admitted to the condenser.

(4) The amount of back pressure or vacuum, whether the reduction is obtained quickly or not, and the amount of compression at the end of the stroke.

It must be borne in mind, however, that as the indicator shows only the pressure at each point of the stroke, the engineer has to account for peculiarities in the form of the diagram by reasoning, and errors are often committed here.

The indicator in a crude form was invented by James Watt. Since his time its construction has been simplified and perfected, McNaught being one of the earliest to effect improvements.

McNaught's indicator.—In the McNaught indicator, which did excellent service with engines fitted with low steam pressures and moving at slow speeds, the pencil is attached directly to the indicator piston, so that their extent of motion is the same. Consequently, they are unsuitable for quick-moving, high-pressure engines, as the necessarily long springs used in them have to be instantly compressed to a considerable extent on the admission of steam, and in quick-moving engines this causes violent oscillation of the pencil and a series of undulations resulting in a serrated diagram which is almost useless as an indication of the action of the steam in the cylinder.

Modern indicators.—To obtain satisfactory diagrams it has been found necessary to fit springs of high tension so as to permit of only a small motion of the piston. This reduces vibration, but to obtain a sufficient height of diagram it necessitates that the motion of the pencil be much greater than that of the indicator piston. The various types of modern indicators differ principally in the means of producing this multiplication, while still keeping the pencil moving in a straight line, and preserving a constant ratio (4 to 6) between the motions of piston and pencil. The difficulty with such motions is not so much to make the pencil move in a straight line, as to insure that it also moves, throughout its range, exactly the same number of times faster than the indicator piston.

The following features are common to all modern indicators, sketches of two of which are given in Figs. 312 and 313. At the lower end of a cylindrical case, A B, is the small steam cylinder in which a piston works practically steamtight, with as little friction as possible. To the lower end of this cylinder a straightway cock, C, is fitted, shown only in Fig. 312, with its end screwed to enable it to be attached to the nozzle of a right-angled three-way cock, called the *indicator cock*, which is connected by pipes to the two ends of the engine cylinder, so that the indicator may be placed in communication with either side of the piston as desired, enabling the two diagrams showing the pressures of the steam on both sides of the piston to be taken on one card. Sometimes the indicator is connected directly to each end of the cylinder, so as to get separate diagrams, which is often desirable in very quick-moving engines, especially

in cases where a fair lead of pipes cannot be readily obtained to enable the two diagrams to be taken on the same sheet.

The upper end, B, of the cylinder is always open to the atmosphere; also, to enable a connection to be made between the under side of the indicator piston and the atmosphere, small holes are made in the shell and plug of the cock C, so that when this cock is shut as regards the supply of steam to the indicator, these small holes open up a connection between the bottom of the indicator and the atmosphere.

A spiral spring of known tension is attached to the piston and also

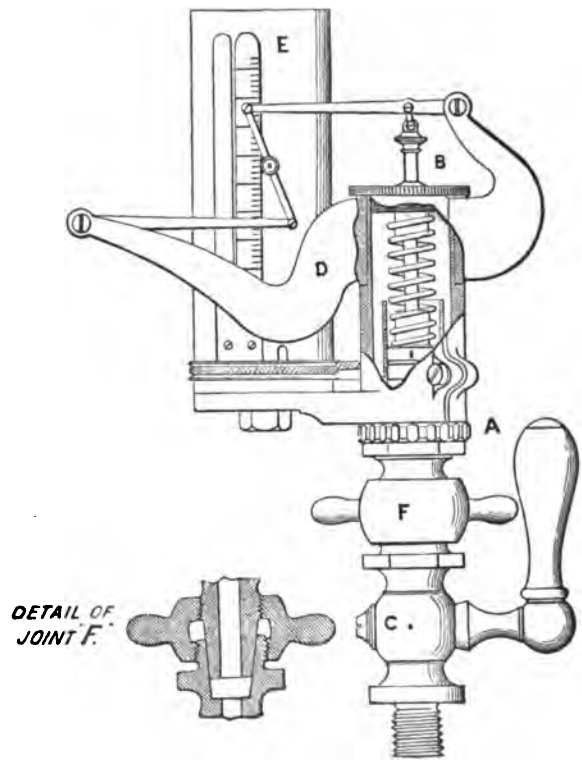


FIG. 311. FIG. 312.

to the top of the casing, and this spring resists the motion of the piston, when acted on by the steam pressure underneath, or by the atmospheric pressure above when the pressure of steam below is less than that of the atmosphere.

The pencil is generally a small brass wire, the paper being specially prepared to enable the wire to mark it. E is a brass drum on which the diagram paper is wrapped and held by clips. This paper cylinder is caused to revolve around a vertical axis by means of a cord deriving its motion from any reciprocating part of the engine that has the same motion as the engine piston, the extent of motion being suitably

reduced by means of levers, &c., to cause the paper cylinder to make about three-fourths of a revolution for each stroke of the engine. The tension is kept on the string, and the paper cylinder brought back to its original position as the engine piston returns, by the action of a spring inside the drum, E. It is very important that the point to which the cord is attached has an exactly corresponding motion to that of the engine piston, but on a smaller scale.

Richards' indicator.—This instrument was the first of the modern indicators with reduced travel of piston to come into use, and its application has for many years been very successful. A sketch of this instrument is given in Fig. 312.

The pencil, instead of being attached directly to the piston-rod, as in the early indicator, is worked by a lever of the third order, the extent of the motion of the piston being only one-fourth that of the pencil, which latter has an extreme travel of $3\frac{1}{8}$ inches. The parallelism of the pencil is maintained by an arrangement of light steel rods, carried by a movable brass bracket, D, fitted on the top of a cylinder. This bracket can be moved round the cylinder by hand, to bring the pencil on or off the metallic paper wound round the barrel.

Crosby's and other 'high-speed' indicators.—Another type of indicator has come into extensive use during the last few years for obtaining improved diagrams from engines of high pressure, working at high rates of speed, such engines as those of torpedo boats, and torpedo-boat destroyers, which often run at over 400 revolutions per minute, with steam pressures of 250 lbs. per square inch in the boilers. It will be readily understood that obtaining satisfactory diagrams from such engines is a work of some difficulty.

In the Richards instrument the rods of the parallel motion are made very light, but it will be seen that there are several rods in the neighbourhood of the pencil, which move up and down through the same distance as the pencil. At very high pressures and speeds, therefore, the momentum of these moving rods produces a slight disturbance in the diagram, and it is important to reduce the weight of the moving parts in the neighbourhood of the pencil as much as possible when the indicator is intended for use with exceptionally high pressures and speeds.

There are several types of indicator in which this is sought to be accomplished, two of which, Crosby's and Darke's, have been used in the Royal Navy for some years, and have given satisfaction. Crosby's high-speed indicator is shown in Fig. 313. In this indicator the motion of the indicator piston is reduced to only one-sixth that of the pencil. The parallel motion and pencil attachment are very light, and the moving parts are few in number, so that there is very little disturbance of the diagram due to the momentum of comparatively heavy moving parts. The spring of the pencil drum is in this indicator a short spiral, instead of a volute spring, as in most other indicators. The upper side of the indicator piston is open to the atmosphere by means of the hole shown. The remaining features of the apparatus will be readily understood from the diagram.

Method of taking indicator diagrams.—The indicator pipes from both ends of the cylinder must first be blown through to clear them from water. The indicator being fixed in position, the string is con-

nected to the indicator lever, and its length adjusted by means of a running loop, to give the proper movement to the paper cylinder. A sheet of prepared paper is stretched smoothly on the paper cylinder, and the ends secured by the spring clips. The indicator cock, and the cock C, are then opened, and the indicator piston allowed to move up and down *till the apparatus is thoroughly warmed* before any part of the diagram is drawn. This will be after a few revolutions of the engine. The handle of the stop cock C should then be turned horizontally to the position in which the bottom of the indicator piston is in communication with the atmosphere.

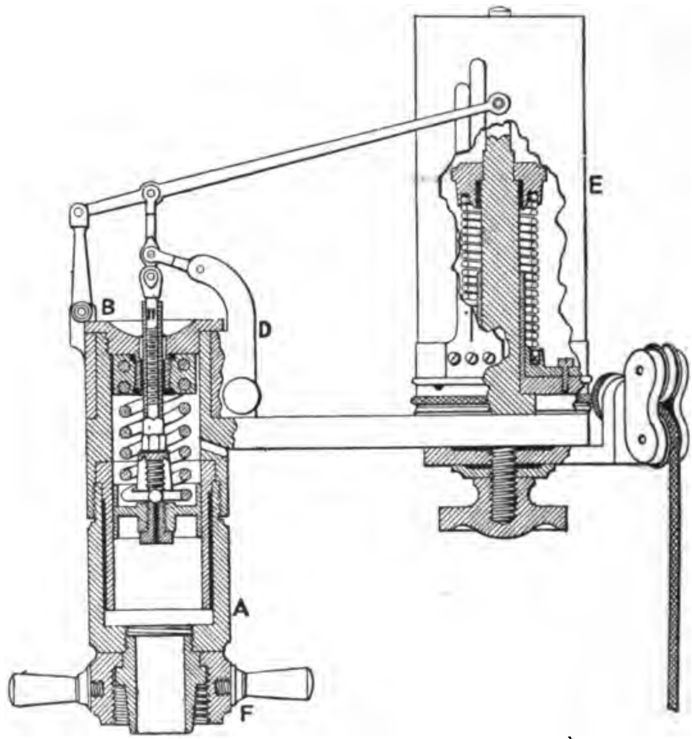


FIG. 313.

In this position the pencil, on being pressed against the revolving paper, will trace a straight line, which will represent the atmospheric pressure, and is called the atmospheric line of the diagram. This is marked C D in Fig. 314.

The cock C on the instrument is then opened for steam, and the indicator cock below it set to make connection with one end of the engine cylinder; the indicator piston will move up and down according to the variation of pressure in the cylinder, and if the pencil be slightly pressed against the paper the combination of the motions of the paper and pencil causes the latter to trace out a curve such as A B D R C

in Fig. 314. A stop is fitted to the revolving arm which prevents the pencil being pressed too heavily against the paper. The pencil traces the part A B Q D during one stroke of the piston, and on the return stroke traces the lower part D R C.

This diagram shows the pressures of the steam on one side of the piston only, during a complete revolution. To ascertain the pressures on the other side of the piston, the indicator cock must be so placed as to open up communication between the indicator piston and the opposite end of the cylinder, when a similar diagram, but reversed, will be obtained. The cocks are then closed, and the diagram removed from the drum.

On the diagrams, besides the time and date, there should be marked the scale of the diagram—i.e. the number of pounds pressure each vertical inch represents—the amount of steam and receiver pressures, and the vacuum shown by the gauges, also the number of revolutions per minute the engine was making at the time the diagram was taken, and the fraction of cut-off of the steam, as shown by the indexes of the links.

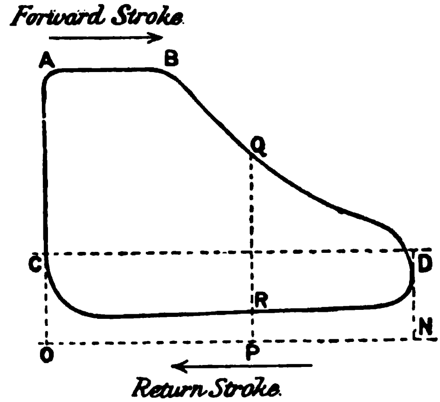


FIG. 314.

Absolute pressure at any point.—If a horizontal line be drawn at a distance below C D, equal, on the given scale, to the atmospheric pressure, say 14·7 pounds per square inch, this line O N will be the *zero line*, or line of no pressure, and all ordinates measured from this base line will represent *absolute pressures* of steam per square inch.

The total length O N of the diagram represents the length of stroke of the piston of the engine, and at the part of the stroke represented by the point P—i.e. when the piston has travelled the fraction $\frac{OP}{ON}$ of its stroke—the forward absolute pressure on the piston will be P Q. When the piston has completed its stroke, and returned to the position P on the return stroke, the absolute pressure on the same side of the piston is represented by P R. This pressure P R is now a back pressure—that is, it resists the motion of the piston.

Mean pressure.—It is clear that the mean of all the distances, such as P Q, will be the mean *forward* pressure during, say, the forward stroke, while the mean of all the distances such as P R will be the mean *back* pressure during the return stroke. The difference between these two pressures—that is, the mean of all the distances, such as Q R, or the mean height of the indicator diagram—is the *mean effective pressure* on this side of the piston during two strokes.

We also have a similar diagram from the other side of the piston, the average height of which gives us the mean effective pressure on the other side during two strokes. It follows, therefore, that the mean of

the average heights of the two diagrams gives us the total mean effective pressure on the piston, to determine which is the principal use of the indicator diagram.

It should be carefully observed, however, that any particular distance, such as Q R, measured from the *same indicator diagram* does not give us the effective pressure on the piston at that point, for the effective pressure at any time is the difference between the forward pressure on one side of the piston and the back pressure *on the other side*. To exhibit, therefore, the forces acting on the piston at each point of the stroke, the forward pressure line of one diagram must be combined with the back pressure line of the other diagram.

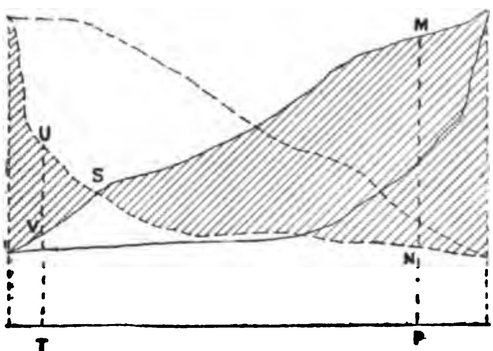


FIG. 315.

Fig. 315 shows this combination. In this figure the diagrams from the two sides are distinguished by full and dotted lines, and the diagram showing the forces acting on the piston at any point of one stroke is shown by the shaded area. For instance, at any point P the resultant pressure on the piston is M N. It will be seen that reversal of the force occurs at the point S, where for an instant no force is acting on the piston, after which, at any point T the resultant pressure on the piston is negative, of amount U V, and tends to bring the piston to rest. The effect of inertia of moving parts is considered in the Appendix.

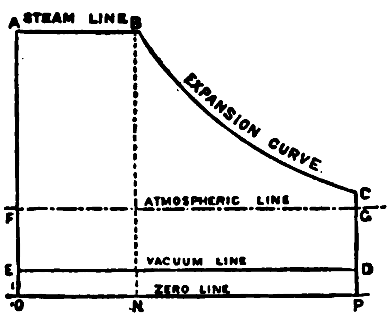


FIG. 316.

Unless these leading principles be well understood erroneous ideas are formed when inspecting an indicator diagram, as regards the distribution of force throughout the stroke.

We will now consider the manner in which the performance of the engine can be deduced from actual indicator diagrams.

Theoretical diagram. — In the first place we will take the theoretical diagram which was explained in Chapter XI., and point out the deviations of actual diagrams from this, resulting from various causes. This theoretical diagram is similar in form to the actual diagram, but much more simple in construction. It is repeated in Fig. 316, and is made up of the admission line, steam line, expansion line, exhaust line, and vacuum line, as indicated in the figure, with the atmospheric line, and the zero or base line, and in its construction it is assumed that (*a*) the full pressure of the boiler steam acts suddenly on

the piston at the beginning of the stroke, and remains constant up to the point of cut-off; (*b*) that the expansion is continued to the end of the stroke; (*c*) that the communication with the condenser is opened at the end of the stroke, the pressure falling suddenly to that in the condenser; and (*d*) that the back pressure remains constant during the whole return stroke.

Form of expansion line.—The exact form of the expansion curve for various conditions of wetness of steam and reception of heat has been discussed in Chapter XII. For most practical purposes, however, the curve is sufficiently accurate when made a common hyperbola—i.e. with the absolute pressure varying inversely as the volume. The following is a useful construction for drawing the hyperbola:—

Let O A represent the absolute pressure of steam and A B the quantity of steam expanding, B being the point of cut-off, O X is the zero line or line of no pressure. Draw B N perpendicular to the zero line. Produce A B and take any point K in it; join O K. From L, where

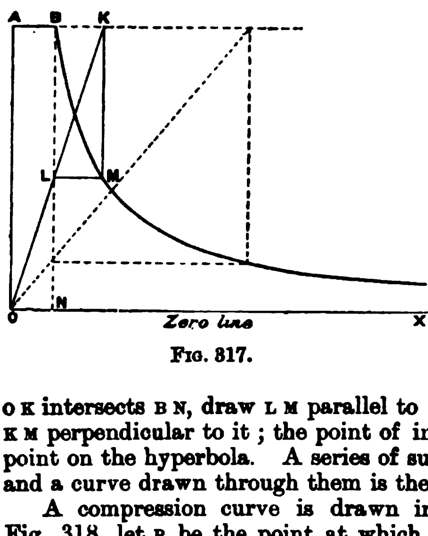


FIG. 317.

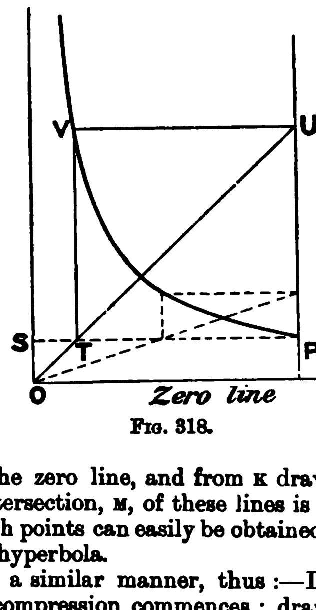


FIG. 318.

O K intersects B N, draw L M parallel to the zero line, and from K draw K M perpendicular to it; the point of intersection, M, of these lines is a point on the hyperbola. A series of such points can easily be obtained, and a curve drawn through them is the hyperbola.

A compression curve is drawn in a similar manner, thus:—In Fig. 318, let P be the point at which compression commences; draw P S parallel to and P U perpendicular to the zero line. Take any point T in P S, and join O T meeting P U in U. Draw U V parallel to and T V perpendicular to the zero line, and their intersection V is a point on the compression curve. A series of such points being found, the curve drawn through them will be a hyperbola, representing approximately the compression of the steam.

We will now point out the deviations from the theoretical diagram that exist in an indicator diagram taken from an actual engine whose various parts are properly fitted and adjusted.

Wire-drawing on steam side.—Wire-drawing is the technical name for the reduction of pressure which steam undergoes by its passage

through contracted areas, and by which its efficiency is reduced. We will consider first the steam side.

If the initial pressure actually shown by the indicator be compared with the boiler pressure, a perceptible difference is always observed. This difference in the cases of high-speed engines with long steam pipes and moderate sized steam ports is often very considerable. A certain difference of pressure must always exist between boilers and cylinders, otherwise the steam would not flow from one to the other. The steam line of an actual diagram therefore necessarily always falls below the boiler pressure line. The admission pressure also generally falls a little as the piston advances and its speed increases, owing to the area of steam ports generally fitted being insufficient to admit steam fast enough to maintain the full pressure. Owing to the very large size which would be required for the slide-valves and cylinder ports of modern high-speed engines in order that no slope of the admission line should take place, a slight sloping of this line is generally allowed at full power in such engines, as this feature is of less importance than the reduction in size of passages and ports, and inertia of the moving slide-valve.

This difference of pressure and sloping of admission line as the stroke proceeds is, therefore, practically necessary to a certain extent, and when not excessive is not to be regarded as a defect. In large quick-moving engines this feature is always noticeable in their full-power diagrams. The difference between the boiler pressure and the mean initial pressure varies considerably with the speed and design of the engine. Even in engines of high pressure and speed it ought not, in cases where reducing valves are not fitted, to exceed 10 per cent. of the absolute boiler pressure.

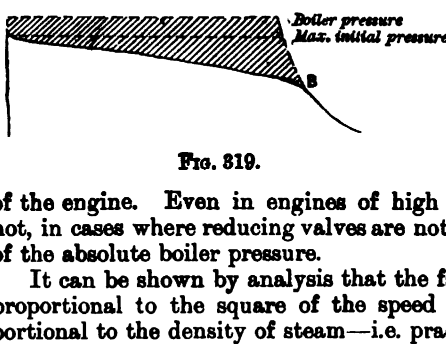


FIG. 319.

It can be shown by analysis that the fall of pressure due to this is proportional to the square of the speed of the piston, and also proportional to the density of steam—i.e. practically to its pressure.

The apparent loss due to this wire-drawing is the shaded area of the diagram in Fig. 319 ; but the actual loss is not so great as this, as the wire-drawing has the effect of drying the steam, so that the heat equivalent of some of this apparently lost work re-appears in the steam. With ordinary slide-valves the cut-off also is not absolutely sudden, as is assumed in the theoretical diagram, but gradual ; so that, instead of having a point at B, the actual diagram would be somewhat rounded, as shown.

Wire-drawing during exhaust.—Again, on the exhaust side, in a simple engine or low-pressure cylinder, as the steam must flow from cylinder to condenser, there must be a difference between the actual back pressure and the pressure in the condenser. This difference is more or less dependent on the freedom of the exit passages and pipes for the steam from the cylinder to the condenser. The difference between the vacuum line of the diagram and that in the condenser is generally from 2 to $2\frac{3}{4}$ lbs., or 4 to $5\frac{1}{2}$ inches of mercury at the maximum power of naval

engines, while at the power for continuous steaming it is about $1\frac{1}{4}$ to $1\frac{1}{2}$ lbs., or 2 to 3 inches of mercury.

To obtain a sufficiently free exhaust the communication with the condenser is opened before the end of the stroke, say at nine-tenths to eleven-twelfths of the stroke, this being necessary to insure the vacuum being nearly complete when the piston commences its return stroke, owing to the exhaust not opening suddenly, but gradually. This early release and gradual opening causes the 'toe' of the diagram to be rounded off, as in Fig. 320 at C D.

Compression or cushioning.—The theoretical diagram assumes that the back pressure remains constant during the whole return stroke. In practice the connection with the exhaust pipe is closed at some point E before the end of the stroke, and the steam then remaining in the cylinder is compressed behind the piston until just before the end of the stroke, its pressure rising to F, when fresh steam enters and a new stroke commences (see Fig. 321).

The imprisoned steam forms a 'cushion,' which tends to bring the piston gradually to rest at the end of the stroke, and greatly reduces the sudden force that would otherwise come on the engine, owing to

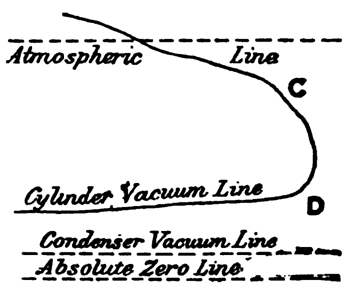


FIG. 320.

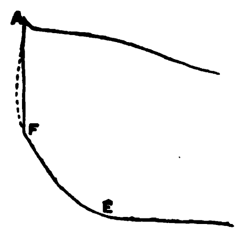


FIG. 321.

the admission of steam of full boiler pressure at the end of the stroke. This action is called 'compression' or 'cushioning,' and has the effect of greatly reducing shocks to the mechanism.

Compression, as we shall see later on, also greatly reduces the loss from clearance. The corner E F of the diagram is called the compression, or cushioning corner.

Pre-admission.—To increase the opening to steam at the beginning of the stroke, which would otherwise be deficient and not permit the steam pressure being maintained sufficiently, the admission of steam occurs at F, just before the completion of the return stroke; the steam pressure then at once rises to the admission pressure, and the compression curve E F gives place to the admission line F A.

Clearance.—In the construction of the theoretical diagram of Fig. 316, it has been assumed that the piston travels the whole of the volume of the cylinder, and that the expansion curve B C represents the expansion of a quantity of steam whose volume is A B, the volume swept out by the piston. Actually, however, not only is there a small space between the piston and the cylinder cover at the end of the stroke, but the passages between the cut-off valve and the cylinder are also

of considerable volume. The sum of these spaces—i.e. the volume of the ports and passages between the piston at the end of its stroke and the slide-valve face—is called the *clearance volume.*

These are spaces through which the piston does not move, but which are filled with steam on admission. The quantity of steam expanding in the cylinder is, therefore, not only that in the volume swept out by the piston, but that in the clearance spaces as well. In actual engines this clearance volume varies considerably ; but usual values for large marine engines are as follows, in fractions of the stroke volume, which is a convenient way of representing it.

Let c = fraction of stroke whose volume is equivalent to the clearance, then

$$c = \frac{\text{volume of clearance space}}{\text{area of piston} \times \text{length of stroke}}.$$

In large marine engines in the Royal Navy, the values of c are usually as follows :—For flat slide valves 12 to 19 per cent., for piston valves 21 to 25 per cent. In the mercantile marine with large engines the clearances vary usually as follows :—For flat slide valves 10 to 15 per cent., for piston slide valves 12 to 20 per cent. In special cases these values are often considerably exceeded ; for example, in one torpedo-boat destroyer the clearance in the H.P. cylinder, with a piston valve, is 34 per cent.

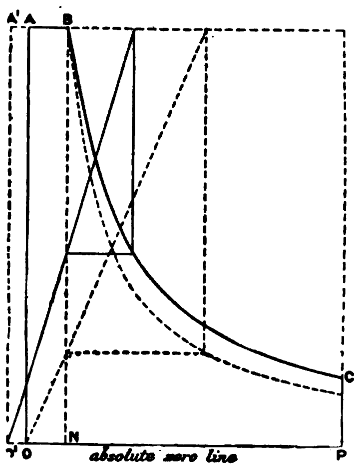


Fig. 322.

Effect on indicator diagram.—The effect of clearance on the indicator diagram is most conveniently represented by measuring off, away from the diagram (Fig. 322), a distance O O′, equal to the length of the stroke the clearance amounts to ; thus $OO' = c. OP$. The quantity of steam expanding is now A′ B, instead of A B, and the hyperbolic curve of expansion must be constructed, as indicated by full lines, by lines radiating from the centre O′. The effect is that the expansion curve lies above that which would be obtained by neglecting the clearance, which in the figure is represented by the dotted lines and curve.

In some old Woolf engines the clearance volume was as great as two-fifths the stroke volume, and the alteration of expansion curve resulting from this was very great. In all marine engines, however, the influence of clearance is considerable, so that it is absolutely essential that the clearance should be known and allowed for, as above, before comparisons of any value can be drawn between the actual expansion curve and the theoretical.

The apparent ratio of expansion before clearance is allowed for, is

$$\frac{OP}{ON} = r' \text{ say } \therefore ON = \frac{OP}{r'}$$

The real ratio of expansion allowing for clearance

$$=\frac{\mathrm{OO}'+\mathrm{OP}}{\mathrm{OO}'+\mathrm{ON}}=\frac{c.\mathrm{OP}+\mathrm{OP}}{c.\mathrm{OP}+\dfrac{\mathrm{OP}}{r'}}=\frac{c+1}{c+\dfrac{1}{r'}}$$

For example, suppose the apparent ratio of expansion of steam in the cylinder, as shown by the cut-off gear, to be 8 times, and the total clearance spaces to be one-eighth of the total capacity of the cylinder, which is not an excessive value :—

Then the actual rate of expansion $r=\dfrac{\frac{1}{8}+1}{\frac{1}{8}+\frac{1}{8}}=4\frac{1}{2}$,

so that the actual rate of expansion of steam will only be $4\frac{1}{2}$ times, instead of 8 times, as indicated by the position of the piston at the point of cut-off.

The effect of clearance is to diminish the efficiency of the expansion and cause waste of steam ; for at the beginning of each stroke all the clearance spaces must be filled with steam, which rushes in from the slide casing and does no work during its admission. In cases of high expansion in a single cylinder, unless special care be taken to reduce the clearance as much as possible, the loss from this cause may become very considerable.

Effect of cushioning on efficiency.—The waste of steam resulting from clearance is reduced by the compression at the end of the return stroke, and if the compression be so great as to raise the pressure in the clearance spaces, just before the point of admission, to the initial pressure of the steam, loss from clearance is practically prevented, as no steam is required to pass from the slide casing into the clearance space. It is evident, however, that the mean pressure of the steam during the stroke is reduced by the considerable amount of cushioning required, and if this amount of cushioning were allowed at full power the size of the cylinders would have to be increased on this account, so that this is not generally done. At lower speeds and powers, with the slide gear linked in, this amount of compression can often be usefully effected. In cases in which engines are worked at very high rates of expansion in a single cylinder, it is always advantageous to use a high degree of compression, not only to reduce loss from clearance, but to prevent shock to the machinery on the change of the stroke by the sudden admission of steam of high pressure to the cylinder.

Faults indicated by diagrams.—The form of diagram obtained from an actual engine in good order and properly adjusted does not differ much from the theoretical diagram, the principal deviations being in the rounding of the corners.

It must be clearly understood that the only fact absolutely given by the indicator diagram is the actual pressure of the steam on the piston during the stroke. In order to draw reliable conclusions from the diagrams, a correct knowledge of the action of the steam in the engine is necessary, and careful study is required to enable the information contained in the diagram to be properly understood.

In dealing with the indicator diagrams of a triple or other compound engine, it should be remembered that the steam line of any cylinder succeeding the high pressure is itself an expansion curve, so

that it must not be expected to be parallel with the atmospheric line; again, the back pressure line of the high pressure and succeeding engines, except the low pressure, also represents the expansion, or compression, of steam in the receiver, often in a complicated manner, so that deviations from the horizontal must often be looked for here.

A few of the more important causes affecting the forms of diagrams are as follows. The diagram taken is generally that of a simple engine, but the corresponding deductions for other diagrams can easily be made.

Steam and exhaust openings too small.—This is the most serious defect that can be shown by a diagram, as it is one that is generally due, not merely to incorrect adjustment of the parts, but to faulty construction, and can only be remedied by enlarging the cylinder ports and passages, which in most cases would involve the substitution of new cylinders.

Fig. 323 is an illustration of the effect on the diagram in this case. On the steam side the insufficiency of steam opening is shown by loss of initial pressure and wire-drawing, while on the exhaust side, in consequence of the restricted area of the exhaust passages, the steam will not be able to escape freely, so that the back pressure will be increased and the vacuum not fully attained until the piston has traversed some portion of its return stroke, the vacuum line sloping downward from D to E instead of being horizontal. It is evident that wire-drawing during the exhaust is more injurious than during admission, as it affects nearly the whole length of the diagram during the return stroke, increasing the average back pressure, and thereby reducing the power of the engine.

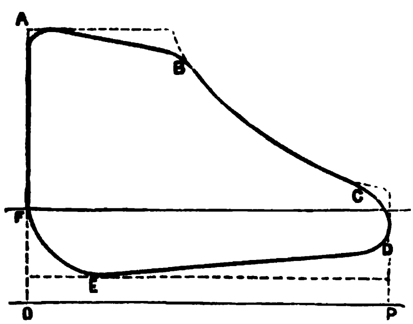


FIG. 323.

Area of cylinder ports and passages.—In the design of engines, the cylinder ports and passages should be made sufficiently large to prevent to as great an extent as possible the fall of pressure during admission. The proper proportion that the area of the ports and passages should bear to that of the cylinder depends upon the steam pressure and piston speed. The following rule is used for calculating the port area:—

$$\frac{\text{area of port}}{\text{area of piston}} = \frac{\text{speed of piston in feet per minute}}{5{,}000 \text{ for H.P.};\ 6{,}500 \text{ for I.P.};\ \text{or } 8{,}000 \text{ for L.P.}}$$

Where possible, however, these figures should be reduced by about 10 per cent. to increase the areas. The area of steam pipe and also the maximum opening for steam is about 70 per cent. of the port area, and the area of exhaust pipe 20 per cent. greater than the port area. It is most important to arrange for the free and unrestricted exhaust of the steam from the cylinder at the end of the stroke, for any defect in this action causes a much more serious reduction of power of the

engines than would arise from deficiency in the area of opening for admission. It is usual to make the exhaust ports and passages at least 50 per cent. greater than the maximum steam openings for this reason.

Suppose the slide-valve incorrectly set on the rod, the eccentric being set at the proper angle of advance.—In this case one end of the valve will have insufficient lap on the steam side and too much on the exhaust, and at the other end of the valve the errors will be of an opposite character. The valve will therefore at one end admit steam too early, continue the admission too long, and cut-off too late, while at the opposite end the operations will be reversed. On the exhaust side, the valve, at the end with early and lengthened admission, will commence to exhaust late and the period of the exhaust will be shortened; at the other end the exhaust will begin early and continue for an increased portion of the stroke. The diagram will therefore be of the character shown in Fig. 324, the faults in the diagrams from *opposite* sides of the piston being of *opposite* natures.

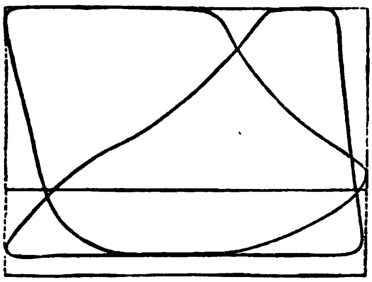


FIG. 324.

Suppose the slide-valve to be correctly set, except that the eccentric is secured in an incorrect position on the shaft.—If the angle of advance be too small, we know that all the actions of the slide-valve for *both* sides of the piston will be too late, the diagram being as shown in Fig. 325, the admission lines sloping inwards and both compression corners being small or non-existent. The late release has the effect that the back pressure does not fall to the condenser pressure till some fraction of the stroke has been performed. This defect is a more serious one than that next described, in which the operations are too early.

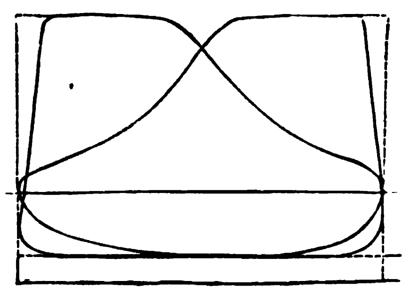


FIG. 325.

If the angle of advance be too great, all the operations on *both* sides of the piston will be too early. The early release causes a considerable fall of pressure before the end of the stroke, and the compression becomes excessive. This form of diagram is approximated to when the steam is worked expansively by the use of the link motion. The effect of 'shortening the link,' as it is technically called, is to shorten the stroke and increase the equivalent angle of advance of the eccentric, by which the actions of admission, cut-off, release, and compression are all made earlier. When not excessive these features of the diagram should not be regarded as defects. Fig. 326 shows an actual diagram from a triple-expansion engine, taken when 'linked in' considerably, from which the effect on the diagram can be seen.

In both these examples the faults in the figures, from the *opposite* sides of the piston, are *similar* in character, and not opposite as in the case of the valve being wrongly placed on the rod. These defects may be remedied by moving the eccentric on the shaft, and setting it with the proper angle of advance, to give the slide-valve the required lead, &c.

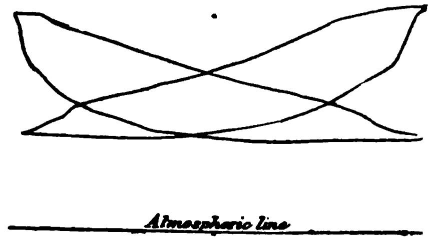


FIG. 326.

Leaky slide-valves and leaky pistons.—If the valves are leaky steam will continue to enter the cylinder after the admission is nominally cut off. In consequence of this the expansion line will rise above the proper or normal curve, the pressure being increased by the leakage during expansion. This would involve a loss of efficiency, and should be guarded against, especially in slow-moving engines, in which the percentage of loss would be the greater. On the other hand, if the piston leak, steam will pass from the steam to the exhaust side of the piston during the stroke, which would cause the pressure during expansion to be lower than it would otherwise have been, the pressure being reduced by the leakage. This is a serious defect, because the steam which passes the piston goes direct to the exhaust, without doing useful work in that cylinder.

In a stage-expansion engine any steam leaking past the piston of a high-pressure or intermediate cylinder does useful work in the succeeding cylinders, so that the loss would not be so great.

The condition of the valves and pistons of an engine as regards leakage is a matter of importance as regards its economy, and insufficient attention is often paid to this question. It is not usually to be detected by a simple inspection of indicator diagrams, as, unless the leakage be excessive, its influence on the form of the diagram is not great.

The proper method to pursue in order to determine whether the pistons and valves are in good condition is to test them by steam pressure, when the engine is at rest, by fixing the piston in some position in which the slide-valve is closed, and applying steam to one side of the piston by means of the starting valves or other means; the leakage of the piston may be observed by the use of an open cock at the other end of the cylinder, such as the indicator cock. By admitting steam to the slide jacket with the valve in the closed position, and opening cocks at each end of the cylinder, the leakage of the slide-valve can be observed. By such means it can be readily ascertained if the pistons and slide-valves are approximately in an efficient condition.

Useful information can be obtained from the indicator diagram by selecting a point about two-thirds along the expansion curve, and ascertaining by construction, by the method described previously, the point on the corresponding hyperbola which has the same pressure as that at a point just after cut-off. The clearance and zero lines

must first be drawn (the amount of clearance in the cylinders of the engines should be known by the engineer of every steam vessel). Vertical and horizontal lines are then drawn through the point P selected, Fig. 327, the former meeting the pressure line at the point near cut-off in Q. Join O Q and complete the rectangle P R; then R is on the hyperbola corresponding to the point P. The relation between the point R and the point S on the actual diagram gives us some information of the kind sought, if it be compared with the similar relation shown on a diagram taken when the engine was new, or known to be in an efficient condition, and when the conditions as regards steam jacketing, pressure, &c., are the same. The correct relation between R and S will depend on the type of engine and whether steam jacketed or not. If R lies beyond the proper position on the outside, leaky slide-valves should be suspected, while if R lies on the other side of the proper position, i.e. towards the vertical axis, a leaky piston is the probable cause, and it should be tested.

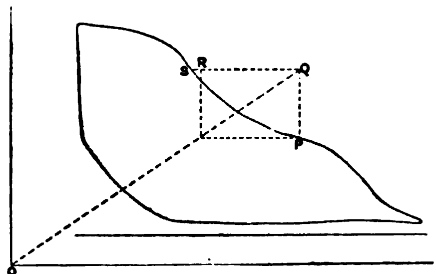


FIG. 327.

In engines with efficient steam jackets and the parts in good condition, the points R and S should not differ by any considerable amount, R generally lying a little outside the diagram. In new engines, if a very considerable difference between these points occurs, investigation will be desirable.

All the defects hitherto discussed influencing the form of the diagram are due to alterations affecting the power and efficiency of the engine. We will now mention a few external causes which affect the form of the diagram only, and which must be carefully avoided if correct inferences as to the action of the steam are to be drawn.

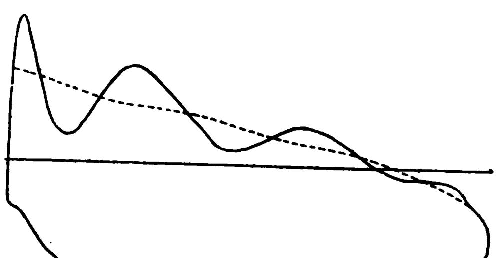


FIG. 328.

Undulation or vibration of the pencil.—This, if it occur, is due to the weakness of the springs, especially when of considerable length. It should be avoided by selecting stronger springs, and indicators having pencil motions suitable to the speed of the engine, as explained previously. If undulations do occur, it will be more correct to calculate the horse-power from a dotted line drawn midway between the crests and hollows, as shown in Fig. 328, than from the actual diagram itself.

Friction of the indicator.—When this occurs it opposes the motion of the indicator piston, and therefore tends to make the indicated

forward pressure less and the indicated back pressure greater than is correct, and so to make the indicated work appear less than is really exerted. In practice, if the instruments are kept in order this will not exist, because the indicator pistons are always made with a slight degree of leakage, so as to make them as nearly as possible frictionless. Sometimes, however, dirty matter is carried into the indicator cylinders with the steam, which would increase the friction, and it is necessary that the instruments should be frequently examined and cleaned to insure correct results being obtained. In cases of high expansion, if the indicator piston be too tight, the defect is sometimes shown by a series of steps on the diagram, the piston, instead of following the steam freely, descending in jumps in consequence of the friction.

This can be tested for one position of indicator piston by drawing the atmospheric line when the pencil has been gradually released, (1) after the indicator piston has been depressed by the finger, (2) after it has been raised by the finger. The two lines should coincide if the condition is satisfactory.

Position of the indicator.—If the position of the indicator is such that a rapid current of steam passes across the nozzle, the steam pressure shown on the diagram will be thereby reduced. Sudden bends, great length and smallness of diameter in the indicator pipes, also tend to reduce the indicated pressure given by the diagrams.

Length of string.—The length of the string should be carefully adjusted before taking the diagram, for if it be either too long or too short the paper cylinder will come to rest before the piston reaches the end of its stroke. The pencil will consequently trace a vertical line when it should be inclined, which will cause the corners to be square and incorrect, and the effect will be the same as if a vertical line were drawn, cutting off a portion of the proper diagram.

Calculation of horse-power from indicator diagrams.—By means of the indicator diagram we are enabled to measure the mean effective pressure on the piston, and consequently, as the area of the piston, length of stroke, and number of revolutions per minute are known, the I.H.P. of the engine can be readily calculated.

If P = mean effective pressure on the piston in lbs. per square inch, and A = net area of piston in square inches:

then $P \times A$ = total pressure on the piston in lbs.

and if L = length of stroke in feet,

N = No. of revolutions of the engine per minute,

then $2\,L \times N$ = speed of piston in feet per minute.

Therefore $P \times A \times 2\,L \times N$ = the number of foot-pounds of work done per minute.

Consequently, since one I.H.P. is equal to 33,000 foot-pounds of work per minute, the I.H.P. developed in the cylinder in question will be

$$\text{I.H.P.} = 2\,\frac{PLAN}{33{,}000}.$$

In this formula A is the mean *net* area of piston. The area of piston-rod must be subtracted from that of the piston, as the pressure does not act on the former area. Similarly with a tail rod, should the

engine be so fitted. In strict accuracy, supposing p_f and a_f the mean effective pressure and net area of the front part of the piston, and p_b and a_b the similar quantities for the back part of the piston, the horse-power will be :—

$$\text{I.H.P.} = \frac{\text{NL}\,(p_f\, a_f + p_b\, a_b)}{33{,}000}.$$

a_f is generally not the same as a_b on account of the area of the piston-rod. They are generally so nearly equal, however, that the error made by substituting $\frac{1}{2}(a_f + a_b)$ for each of them is inappreciable. This is what is done to arrive at the simple and usual formula :—

$$\text{I.H.P.} = \frac{2\,\text{PLAN}}{33{,}000}.$$

In this expression, P and N are the only variables for the same cylinder. It is therefore usual in practice to combine the constant quantities to further facilitate calculation. The *cylinder constant* for any engine is clearly

$$= 2\,\frac{\text{L A}}{33{,}000} = \text{C say}$$

A being the mean net area of piston.

This constant, multiplied by the mean pressure calculated from the diagram and by the revolutions of the engine per minute, will give the I.H.P. of the cylinder,

$$\text{or, I.H.P.} = \text{C} \times \text{P} \times \text{N}.$$

If there be more than one cylinder, the powers developed in the several cylinders must be added together to obtain the total I.H.P., care being taken to remember that the scales used in the cylinders of a stage-expansion engine vary with the pressure.

Calculation of mean effective pressure from the indicator diagram.—The following is the method commonly adopted for ascertaining the mean effective pressure from a pair of indicator diagrams.

The total length of the diagram is divided into ten equal parts, and vertical ordinates are drawn at the middle points of the spaces thus formed. The first and last ordinates will then each be $\frac{1}{20}$th of the length of the diagram from the end, and the common distance between the several ordinates will be equal to $\frac{1}{10}$th of the length of the diagram. This method of division is shown on the diagrams of Fig. 334.

Diagrams are taken from *each* side of the piston, and the lengths of the several ordinates, intercepted between the forward and back pressure lines of one diagram, are measured on the required scale, added together, and divided by 10. The same process is carried out on the other diagram, and the mean of the two means thus obtained gives the *mean effective pressure* for the complete double stroke of the engine, which is used in calculating the I.H.P.

To facilitate calculation, the lengths of the ordinates are usually measured successively on a strip of paper, the second ordinate commencing at the end of the first, and so on. The total length thus obtained is measured on a scale made ten times as great as the scale of the diagram, so that the mean pressure may be read off at once.

The other method used for ascertaining the mean effective pressure is by the use of the 'planimeter.'

Calculation of steam used as shown by the indicator diagram.— From the indicator diagram the quantity of steam, and consequently of water, used per I.H.P. per hour, as accounted for by the indicator diagram, may be ascertained. The values calculated from the diagrams are not, however, representative of the actual amount of water passing through the engine, but are always less than the actual quantities of steam used, and often very considerably so, because they do not include the waste of steam due to liquefaction in the cylinders, radiation, and other causes. Relatively, however, as showing the differences in the performance of similar engines, or of the same engine working under different circumstances, this application of the diagram is of value, and some information may be gained from such calculations, provided their real meaning and limitations be clearly understood.

From what has been said before it is clear that the working steam in the cylinder consists of two parts, viz. (*a*) the portion that passes through the engine at each stroke, being received from the slide casing at the beginning of the stroke and exhausted from the cylinder during the return stroke, and (*b*) the cushion steam, viz. that part of

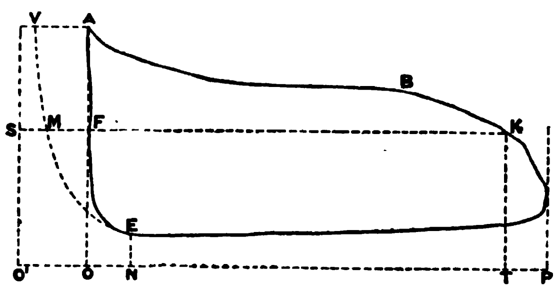


Fig. 329.

the steam which is present in the clearance spaces and cylinder when compression takes place, and which is retained in the cylinder and compressed till the end of the stroke. This volume of steam, therefore, does not leave the cylinder at all, but is alternately compressed and expanded at each stroke.

The consumption of steam does not include this cushion steam. We have, therefore, to ascertain the quantity of steam that *leaves the cylinder* during each stroke. To do this we must select a point, say K, Fig. 329, on the expansion curve in such position that it is undoubtedly before release has taken place; then clearly at the point K in the expansion the total volume of steam present is S K.

Let E be a point on the back pressure line, just after the exhaust has closed and compression begun, then the quantity of steam of volume O′ N and pressure N E is retained in the cylinder and does not leave it. This quantity must therefore be deducted from the total quantity just before release, viz. the volume S K of pressure T K, to obtain the quantity leaving the cylinder and used per stroke. To reduce these quantities to the same pressure we draw a saturation curve through the point E meeting the horizontal line through K in M.

Then S M will be the volume of the cushion steam when its pressure is K T, which steam does not leave the cylinder, therefore the difference—i.e. the volume represented by M K—on the same scale that O P represents the stroke volume, is the amount of steam that leaves the cylinder at each stroke, so that this quantity, at pressure K T is the consumption of steam per stroke as accounted for by the indicator diagram. This quantity may be termed the *working steam*, while the remainder, viz. S M, is termed the *cushion steam*.

Theoretical diagram of a stage-expansion engine.—It was explained in Chapter XVIII. that the total expansion of steam successively in the cylinders of a compound or multiple-expansion engine is theoretically the same as if it had been carried out entirely in the low-pressure cylinder only. We will take first the simplest possible case, in which there is no clearance and no wire-drawing, and that there are receivers between the cylinders so large that the pressure in them remains constant, therefore the back pressure in the smaller cylinder, and admission pressure in the larger cylinder of any pair, remain constant and equal to the receiver pressure. Let us further assume that the expansion in each cylinder reduces the pressure to the back pressure in the cylinder, and therefore to the admission pressure in the succeeding one.

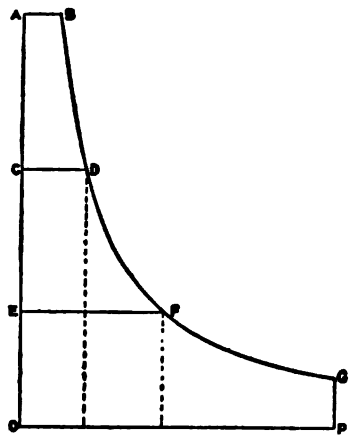


FIG. 330.

It is clear that in any multiple-expansion engine, when it has been running sufficiently long to have assumed a steady condition, *the quantity of steam entering and leaving any cylinder is the same throughout.* In actual engines, of course, part of this steam exists in the form of water. Let A B D C be the high-pressure diagram of this engine (see Fig. 330), then the volume C D of steam at the receiver pressure O C leaves the high-pressure cylinder and enters the receiver; also, as the same quantity of steam enters the intermediate cylinder up to the point of cut-off, therefore C D is the admission line of the intermediate-pressure diagram, and its expansion curve, as the quantity of steam expanding is the same, will be D F, a continuation of the hyperbola B D. The intermediate-pressure diagram is therefore C D F E, where E F is the volume of the intermediate cylinder. Similarly, E F to P O is the low-pressure diagram where O P is the final volume of steam—i.e. the volume of the low-pressure cylinder. The expansion curve, therefore, is exactly the same as if the volume of steam A B of pressure O A had been expanded in a single cylinder of volume O P.

This assumption of a large receiver and constant receiver pressure is one extreme case; the assumption of no reservoir at all is another extreme case, which is dealt with at the beginning of Chapter XXXIII.

With the proportions of cylinders necessary from practical considerations, the powers developed in the various cylinders would be very

unequal, that in the low pressure greatly preponderating, while the necessary cut-offs would be inconvenient.

To obtain greater equality in the powers developed in each stage it is necessary to lower both receiver pressures, which is done by making the cut-off in the low-pressure and intermediate-pressure cylinders later than before. A greater volume of steam is therefore drawn from the receivers per stroke, so that as the total weight of steam passing into the other cylinders must still be equal to that discharged from the high-pressure cylinder, the pressure gradually falls in the receiver till the pressure is such that the volume being taken by the receiving cylinders equals the quantity discharged from the high-pressure cylinder.

Let us suppose the cut-off in the intermediate cylinder, Fig. 331, altered so that the volume of steam admitted is increased from C D to C′ D′; then C′ D′ being the volume of steam admitted, as its weight remains unaltered its pressure must be O C′, and C′ D′ will be the steam line of the intermediate-pressure diagram. The pressure at release in the high-

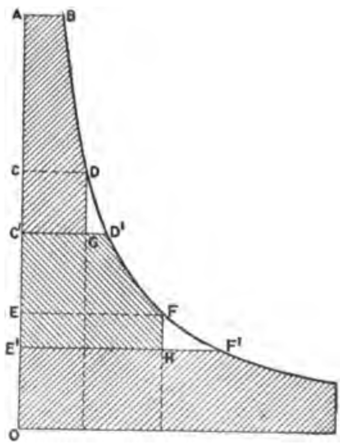


FIG. 331.

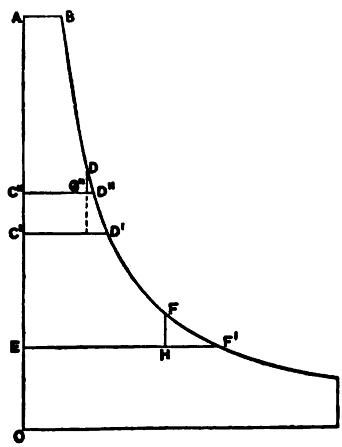


FIG. 332.

pressure diagram, viz. O C, remains unaltered, and is now greater than O C′, the pressure in the receiver, so that on release from the high-pressure cylinder a sudden drop of pressure occurs represented by D G. The triangular area D G D′ therefore disappears, and is mostly lost. A similar action takes place between the intermediate pressure and the low pressure, and the sketch shows the effect of making the cut-off in low-pressure cylinder later—viz. from E F to E′ F′.

Effect of altering the cut-off in the intermediate or low-pressure cylinder.—The preceding reasoning enables us to understand the effect of altering the cut-off in the various cylinders of a triple-expansion engine on the distribution of the power. Suppose we consider the intermediate-pressure diagram C′D′ F H E, Fig. 332. Now if we cut off earlier in this cylinder, as the *weight of steam passing through it remains the same*, its initial pressure must be increased. If the cut-off is made earlier, so that the volume admitted is reduced from C′ D′ to C″ D″, the pressure of steam in the receiver will be altered from O C′ to O C″ : this

will have the effect of increasing the power developed in the intermediate-pressure cylinder and reducing that developed in the high-pressure cylinder, the total power remaining practically constant. The intermediate diagram will be increased to C″ D″ F H E, and the high pressure reduced to A B D G″ C″. Similarly, making the cut-off later in the intermediate-pressure or low-pressure cylinder of a triple-expansion engine has the effect of reducing the work done in that cylinder and increasing the work done in the preceding cylinder, the total work remaining practically unaltered.

Making the cut-off later in the high-pressure cylinder has, of course, the effect of increasing the total power of the engines. In this case the work done in *each* of the cylinders is increased, as the quantity of steam passing through the engines is increased. Similarly, if the cut-off is made earlier the power is reduced throughout.

In a naval engine, owing to the proportions that have to be adopted, although the cut-offs at full power are comparatively late in all cylinders, the high-pressure cylinder still does less and the low-pressure more than its proportion of the work. At reduced powers the cut-offs are made earlier in all cylinders, so that the proportion of power gradually increases in the high-pressure cylinder, and is gradually reduced in the low-pressure, till at very low powers the high-pressure does more than its proportion, and the low-pressure cylinder less.

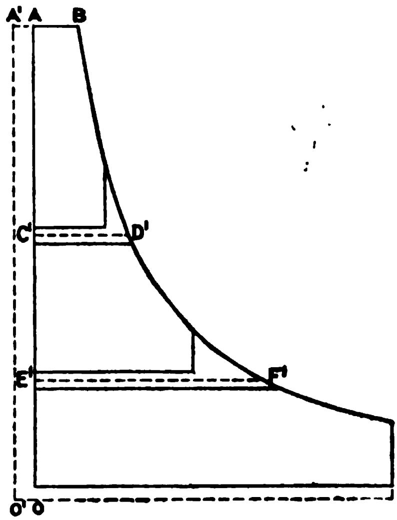


FIG. 333.

The independent linking-up gear fitted to most vessels enables the slide-valves of the low-pressure and intermediate cylinders to be linked up more than the high-pressure when so working, so that if required the receiver pressures can be raised, and the proportion of power developed in their cylinders be brought to nearer an equality with that being developed in the high-pressure cylinder.

These diagrams enable us to clearly see the effect of all such alterations of relative cut-off in the various cylinders on the distribution of the power.

Theoretical diagram taking account of clearance and wire-drawing between cylinders.—Clearance greatly affects the preceding calculations, and is a very complicated factor. We will consider here the case of an engine in which the clearance volume is the same in all cylinders, with no compression. The construction is then the same, except that the hyperbolic expansion curve is drawn with axis O′, instead of O, O O′ being the clearance volume, as shown in Fig. 333.

The effect of wire-drawing between the cylinders is to raise the back

pressure line of the high-pressure diagram, and to depress that of the intermediate-pressure diagram a little above or below the receiver pressure O C′, and similarly for the lines between the intermediate-pressure and low-pressure cylinders.

Approximate theoretical indicator diagrams from an engine.—Proceeding as above, rounding off the corners, and allowing a little excess back pressure beyond C′ D′ and initial pressure in the intermediate-pressure cylinder below C′ D′ of amounts depending on the type of engine, and obtained from experience, we are enabled to roughly approximate to the indicator diagrams to be expected from a given engine.

Having dealt with the theoretical diagram we proceed to consider the case of indicator diagrams from actual engines, and the process of combining them to obtain as much information as possible.

Process of combining diagrams of actual stage-expansion engines.—The scales of pressures and volumes of the indicator diagrams obtained from any compound engine differ for the various cylinders, so that it is customary for various purposes to combine all the diagrams, so as to exhibit on the same scale and on one diagram the changes of pressure and volume undergone by the steam during its passage through all the cylinders. Such a diagram may also roughly exhibit the effect produced with that which would be theoretically obtained by the steam expanding in a single cylinder.

The diagram may be combined in various ways, depending on the purpose for which it is required. In every method the effect of clearance is absolutely necessary to be taken account of to obtain even an approximate idea of the action taking place. In stage-expansion engines certain large sources of loss appertaining to simple engines, losses not shown by the diagrams, are avoided, while other losses are introduced, consisting of the losses between the cylinders due to sudden expansion, friction, and wire-drawing, which it is most important to reduce to the smallest possible extent. The process of combination of diagrams gives graphically an approximate idea of these losses, and enables us to study their further reduction.

Usual method.—The usual method of combination is as follows. The diagrams when combined must be of such relative length as to represent the stroke volumes of their respective cylinders, while the scale of pressures must be identical.

A scale of volumes and pressures having been decided upon, each of the diagrams to be combined is divided into a certain number of equal lengths, and ordinates are erected at the middle of each of these divisions. The low-pressure stroke volume is set off on the zero line to the agreed scale, and the atmospheric line drawn above it on the proper scale of pressures. A distance O O′ is then set off equal to the clearance volume of the low-pressure cylinder, as shown in Fig. 334. The distance O P is then divided into the same number of equal parts as the original diagram, and ordinates erected at the centre of each as before. The several ordinates of the original low-pressure diagram are then carefully measured, and transferred on the new scale to the corresponding ordinate on the combined diagram, and curves drawn through the ends of the ordinates to represent as closely as possible the original diagram.

The intermediate diagram is then served in a similar manner, the end of its diagram being placed at M, such that L M is equal to the clearance volume of the intermediate cylinder on the agreed scale of volumes, and M N, representing the stroke volume of the intermediate cylinder, is then divided as before, and the intermediate diagram constructed. Similarly for the high-pressure diagram. The various distances A C, L M, and O′O represent the clearance volumes of the respective cylinders. We have now a combined diagram such that at any point, say V, the pressure of steam and volume occupied are represented by V T and V S. A hyperbola or saturation curve is generally drawn through the estimated point of cut-off to complete the figure.

Such a diagram gives, when comparing on the same basis engines of different design, an approximate idea of the amount of wire-drawing, and losses between the cylinders. For design work also the proportion between the area of the actual indicator diagrams and that of the hyperbolic enclosure gives us a fraction known as the *design factor*, which becomes useful in estimating the size of cylinders required for similar engines.

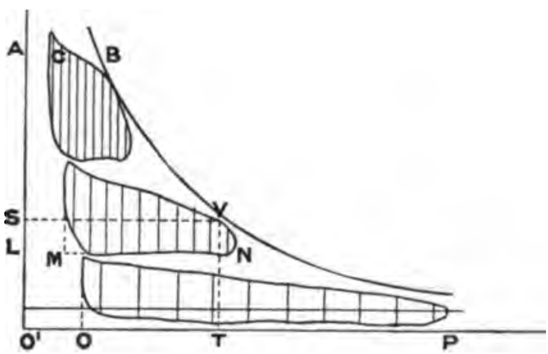


FIG. 334.

A set of indicator diagrams taken from the engines of H.M.S. 'Powerful' are given in Fig. 335, while the combination of these diagrams is shown in Fig. 336. It will be seen that their expansion curve, if drawn to touch the three diagrams, falls considerably below the hyperbola. The spaces between the three diagrams represent approximately the losses due to the resistance of the passages between the cylinders, while the spaces between the release lines and the dotted expansion curve represent approximately the losses from sudden expansion on admission to the receivers.

Referring to Fig. 336 it must be observed that, owing to the varying amounts of clearance volumes, the quantity of steam expanding in the three cylinders is different, so that one continuous curve, either hyperbolic or saturation, cannot really represent the expansion curves of the three diagrams, but, provided its limitations are understood, useful information may still be obtained from such a diagram. To study the form of the expansion curves more accurately, a hyperbola should be drawn for each diagram through its point of cut-off.

It is often found that the diagrams when combined in this way

overlap one another, and there is nothing wrong in this being so, for any given point in the steam line of the intermediate diagram, for instance, does not correspond with the point on the high-pressure back-pressure line on the same ordinate. The point of correspondence must be obtained by calculation, knowing the relative position of the pistons from the arrangement and sequence of the cranks.

Combination to exhibit the real losses of pressure by wire-drawing between cylinders.—A useful diagram may be obtained by setting out the back pressures of one diagram, when in connection with the receiver, and on the same ordinates the forward pressure of the succeeding diagram *at the same instant.* A convenient plan is to do this in such a manner that the abscissæ represent the number of degrees one of the cranks has travelled through, while the ordinates represent the pressures in the cylinders when they are in connection with the

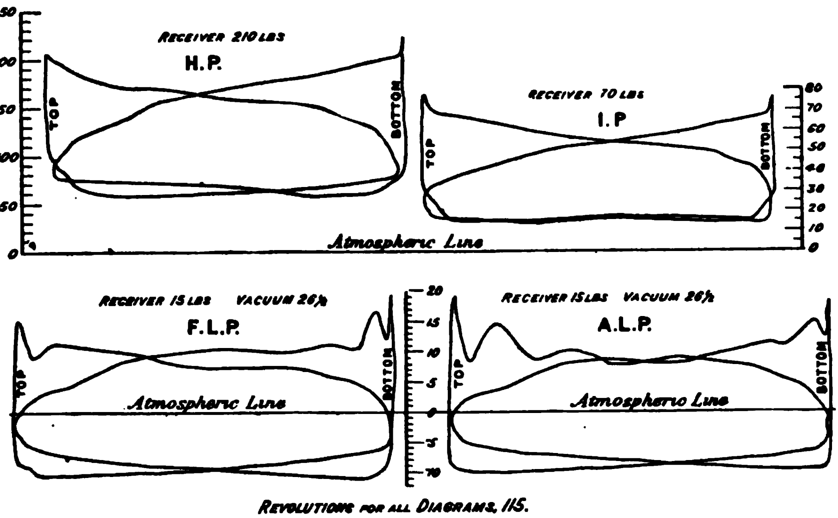


FIG. 335.

receivers. The parts of the diagrams to be so treated would be those between release and compression in the high, and between admission and cut-off in the intermediate, and similarly for the back pressure of the intermediate and the steam line of the low. Sometimes the entire diagrams are thus transformed, but little useful information can be extracted except from the parts mentioned. When these parts are thus transformed, the overlapping previously referred to will be found to have disappeared, and the loss of pressure by wire-drawing at any point is seen.

Fig. 338 shows this combination for the diagrams of H.M.S. 'Powerful' previously dealt with. The full lines in each case represent the pressures in each cylinder when in connection with the receiver, the dotted lines representing the pressures when the cylinder is not in connection with the receiver. The actual loss by wire-drawing can be

seen at any point. It will be useful for students to draw such a diagram for themselves from a given set of indicator cards.

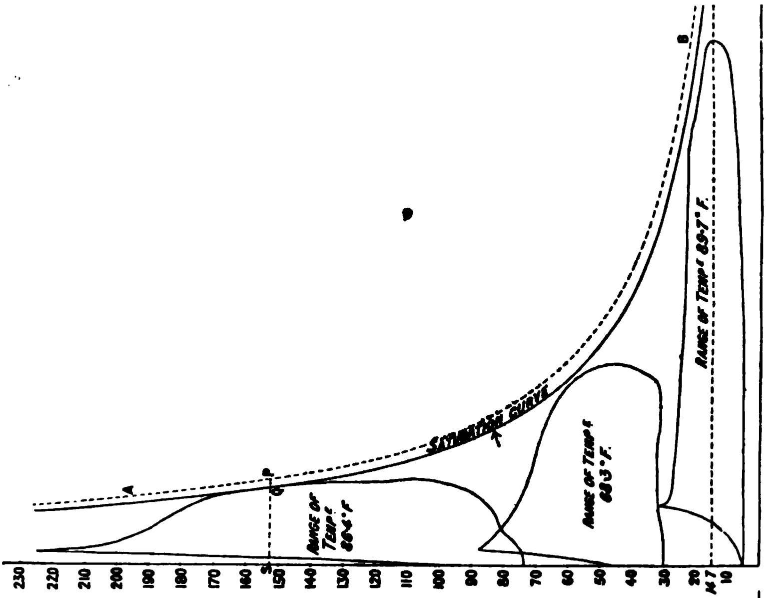


Fig. 337.

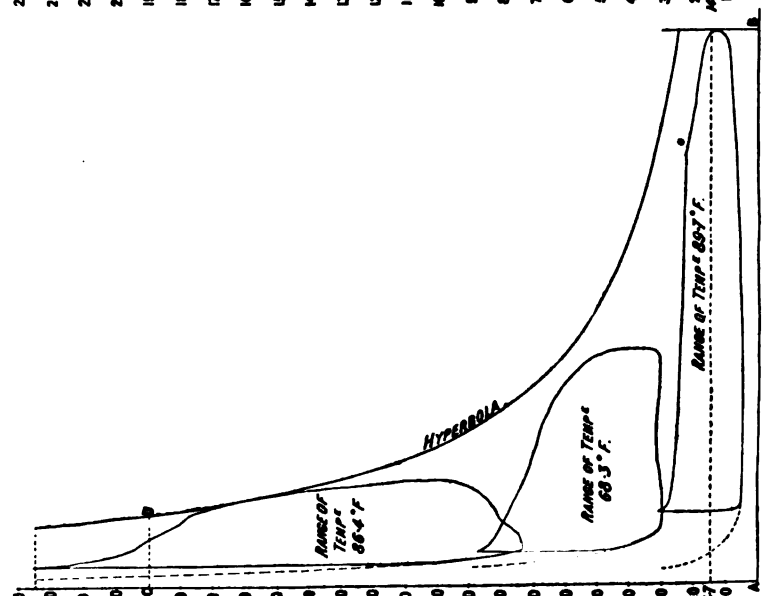


Fig. 336.

Combination to exhibit the distribution of water in the cylinders.— The preceding combined diagram indicates the volume and pressure

of the total amount of steam in the various cylinders, which, as previously explained, is of varying quantity, owing to the different amounts of the cushion steam. An important plan of combining the diagrams is to convert them in such a way that this varying cushion steam is eliminated, so that the diagrams will represent the volume and pressure of the 'working steam'—i.e. the weight of steam discharged from each cylinder per stroke—which amount is constant for all the cylinders. The saturation curve will then be the same for all the diagrams. To do this it is necessary to subtract from the volume of steam at each pressure on the diagram a quantity equal to the volume the cushion steam would occupy at that pressure. The method already described must be pursued to obtain this, the saturation curve E M, of Fig. 329, being continued to the initial pressure. The diagrams are then redrawn by setting off from the line of zero volume, abscissæ equal to the horizontal distances of the indicator diagram from

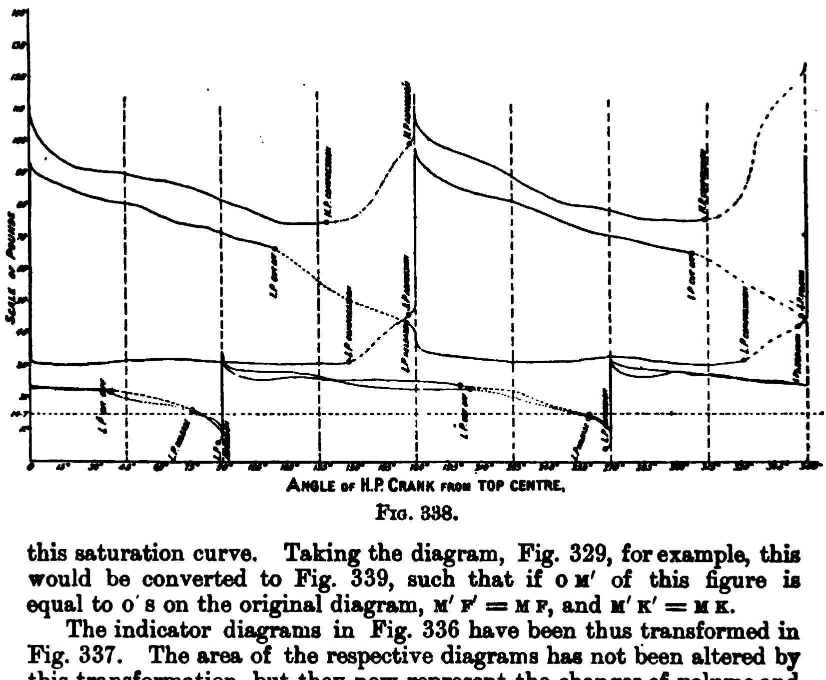


Fig. 338.

this saturation curve. Taking the diagram, Fig. 329, for example, this would be converted to Fig. 339, such that if O M′ of this figure is equal to O S on the original diagram, M′ F′ = M F, and M′ K′ = M K.

The indicator diagrams in Fig. 336 have been thus transformed in Fig. 337. The area of the respective diagrams has not been altered by this transformation, but they now represent the changes of volume and pressure in all the cylinders of the constant quantity of *working steam*. To obtain the full advantage from the study of such a diagram, the quantity of feed-water used in the cylinders per stroke must be known. If this be known, a saturation curve for that quantity of steam is drawn, as A B in Fig. 337. If any horizontal line, such as S P, be now drawn, when the steam is shut off from the receivers, S Q will represent the volume of steam present in the cylinder and Q P the volume of steam condensed in the form of water.

Generally, however, the quantity of water used is not known, but

useful information may still be obtained from the diagram by drawing a saturation curve which just touches the diagram, This will then be a *curve of uniform wetness*, equal to the smallest wetness shown by the diagrams, and the relative liquefaction as the steam passes through the various cylinders will be indicated by the relation of the diagram to this curve.

This curve is drawn for the combined diagrams of H.M.S. 'Powerful' in Fig. 337, from which it will be seen that the wetness of the

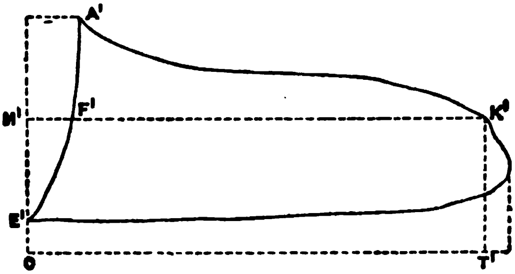


FIG. 339.

steam is greater in the I.P. and L.P. cylinders than in the H.P., but the difference is not great. The pressures maintained in the cylinder jackets influence this. On this occasion the jacket pressures were lower than the initial pressures in the various cylinders, and had higher pressures been maintained, the converted diagrams would probably all have touched the saturation curve, indicating a constant amount of moisture present in each cylinder.

CHAPTER XXVIA.

THE TORSION METER.

No instrument analogous to the indicator as applied to reciprocating steam engines has yet been devised for measuring the horse-power developed in turbine installations. To obtain the power in such cases an instrument called a torsion meter, which measures the torsional movement of the propeller shafts, is used.

Considering a shaft of circular section composed of an elastic material, to which a twisting moment is applied—we know that up to a certain limit the twisting moment applied bears a constant relation to the resulting angular distortion of shaft, and this relation is expressed by the formula

$$C = \frac{12\,F \times 12\,L}{Ia \times \theta},$$

or

$$F = \frac{C \times Ia \times \theta}{144\,L}.$$

C in lbs. per square inch is termed the co-efficient of rigidity, and, up to the elastic limit, is a constant for the shaft under consideration. F = torsion or turning moment on shaft in foot lbs.; θ is the angular distortion, in circular measure, between the two points on shaft which were originally in the same straight line parallel to the shaft axis, and distant L feet from one another, while Ia is the geometrical moment of inertia of the shaft cross section in inch units, which can be calculated from the dimensions of the shaft from the formula $I_a = \frac{\pi}{32}(d_1^4 - d_2^4)$, where d_1 and d_2 are the external and internal diameters of shaft respectively. If the shaft is a solid one, $d_2 = 0$, and $I_a = \frac{\pi}{32}d_1^4$.

Formula for shaft horse-power.—It is evident from this formula that if we have the means of measuring the angle θ or some ratio of it, then knowing the co-efficient C for the material of which the shaft is made and the dimensions of the shaft, we can deduce the value of F which produced this distortion. Having obtained the value of F, the horse-power is deduced from the formula :—

$$\text{S.H.P.} = \frac{2\,\pi\,N}{33000} \times F,$$

in which N represents the revolutions per minute obtained by a counter or other means, and hence :—

$$\text{S.H.P.} = \frac{2\,\pi\,\text{N}}{33000} \times \frac{\text{CI}a\,\theta}{144\,\text{L}}.$$

Relation to indicated horse-power.—It should be carefully noted that the power obtained in this manner does not correspond to the indicated horse-power developed in the cylinders of a reciprocating engine, but to the brake horse-power. To obtain the corresponding indicated power of a reciprocating engine, the torsion meter-power must be divided by the estimated mechanical efficiency of such an engine, so that a larger value will be obtained. In comparing the indicated horse-power of a reciprocating engine with the torsion meter-power obtained from the turbine installation of, say, a sister-ship, this difference must be taken account of.

The torsion meter has also been usefully applied to ascertain the brake horse-power delivered by large reciprocating steam engines in which the actual measurement by a brake would be practically impossible, and from this and the indicated power the efficiency of the mechanism of the engines is readily deduced. The values obtained for efficiencies in Chapter II. have been thus obtained.

Experimental determination of deflection.—The distortions for the various turning moments which obtain in propeller shafting are of necessity very small, and as a consequence their measurement cannot be accomplished by ordinary methods. Various plans of measuring the angular distortion of shafts have been proposed, and several have been tried; but, as these instruments are still not fully developed, it is probable the best form has yet to be evolved. Two examples are given hereafter. In one—Denny & Johnson's—the measurement is effected by means of electrical appliances, and in the other—Bevis & Gibson's—by means of a beam of light.

It is evident that the success of the torsion meter will be dependent in no small measure on the value assumed for the co-efficient of rigidity, on its constancy for the range of power transmitted by the shaft, and on the accuracy of the value of Ia assumed, the latter being somewhat difficult to calculate in actual practice owing to the presence of couplings, small variations in diameter of shaft, &c. In many cases a value has been assumed for the co-efficient of rigidity which is based on the results obtained from a number of calibrations of iron and steel shafts of different dimensions, and for ordinary purposes it will be sufficient to take such a value and make an approximate calculation of Ia, but in cases where great accuracy is desired, and especially for hollow shafting, and also to make due allowance for the couplings, the actual shafts to which the instruments are applied are statically calibrated. The following is the method recently adopted in the case of a shaft made up of two lengths of dimensions $10\frac{1}{2}$ ins. and $6\frac{1}{4}$ ins. external and internal diameters respectively, bolted together by couplings, the aggregate length being 46 feet.

The shaft was carefully supported in its bearings carried on the shop floor and anchored at one end to angle brackets, which were rigidly secured to the shop foundations and ballasted by a weight of 20 tons. Rigidly secured to the free end of the shaft was a beam, and

on this beam and distant 8 feet from the shaft centre was secured a box in which variable weights were deposited. Simultaneous readings for each load were taken at a radius of 8 feet by two double-ended pointers secured to the shaft 43 feet apart at the positions of the torsion meter wheels when in the ship, the axes of the pointers lying on a diameter of the shaft. The ends of these pointers passed over graduated arcs arranged so as to be concentric with the shaft, and this enabled a sensible movement to be registered with very small angular distortions.

Weights varying from $\frac{1}{2}$ ton up to $5\frac{1}{4}$ tons, so arranged as to maintain the centre of gravity of the load at a distance of exactly 8 feet from the shaft centre for all the various loads, were successively placed in the weight box and the readings of the indexes corresponding to each load were carefully noted by the pointer moving over a fixed scale with vernier and the differences recorded; similar readings were also taken when unloading the weights.

The distortions thus obtained when plotted, with turning moments measured along the base and distortions measured vertically, practically fell on a straight line passing through the origin, thus showing that, for the range of turning moment exerted, which covered that corresponding to the horse-power likely to be transmitted by shaft, the distortions varied directly as the turning moment, or $F = \text{constant} \times \delta$, where F = turning moment in foot lbs. and δ = distortion in inches.

In addition to the readings taken at the end of shaft at a radius of 8 feet, others were taken by micrometer gauges at points at the shaft itself at different distances from the anchored end; these readings, corrected for length and radius, agreed closely with those obtained at the end of shaft. Three tests were made on the shaft, and the mean of the mean distortions for all loads was taken to obtain the constant in the horse-power formula as follows:—

For a turning moment of 8 foot tons the mean of means of the differences between the distortions at the two indexes in the three experiments was 0·7375 inches at a radius of 8 feet, and substituting these experimental results in the formula, $F = \text{constant} \times \delta$ where δ = the above distorsion in inches at radius of 8 feet, the value of the constant is $= \frac{8 \times 2240}{\cdot 7375} = 24298$ and the formula becomes $F = 24298 \times \delta$.

Now, applying the formula $\text{S.H.P.} = \frac{2\pi N}{33000} \times F$, where N = number of revolutions per minute, and substituting the value for F obtained above, we have

$$\text{S.H.P.} = \frac{2\pi N}{33000} \times 24298 \times \delta = K \times N \times \delta = 4{\cdot}626 \times N \times \delta,$$

which will give us the horse-power transmitted by the shaft, N being obtained by the counter and δ by the torsion meter.

Before any readings are taken the shaft is tested as to the alignment of the bearings and is twisted several times through the full range of torque by means of the ordinary loading weights in order to minimise any initial internal stresses and to get the shaft into an elastic condition; the shaft is jarred by mallets before taking each reading in order to assist in eliminating the effect of statical friction at the bearings.

Description of apparatus.—Figure 339A shows a general outline of the Denny & Johnson torsion meter, and the following is a brief

description :—Two light gun-metal wheels or pulleys A and B are secured to the shaft at a suitable distance apart; the distance is made as large as possible, within the limits of registration of the standard instrument, the object being to get a large distortion when transmitting low powers, so that under these conditions the distortion and therefore the horse-power may be accurately determined. A permanent bar magnet M is recessed into each wheel as shown in figure and pointed at the projecting end; preferably above each wheel and rigidly

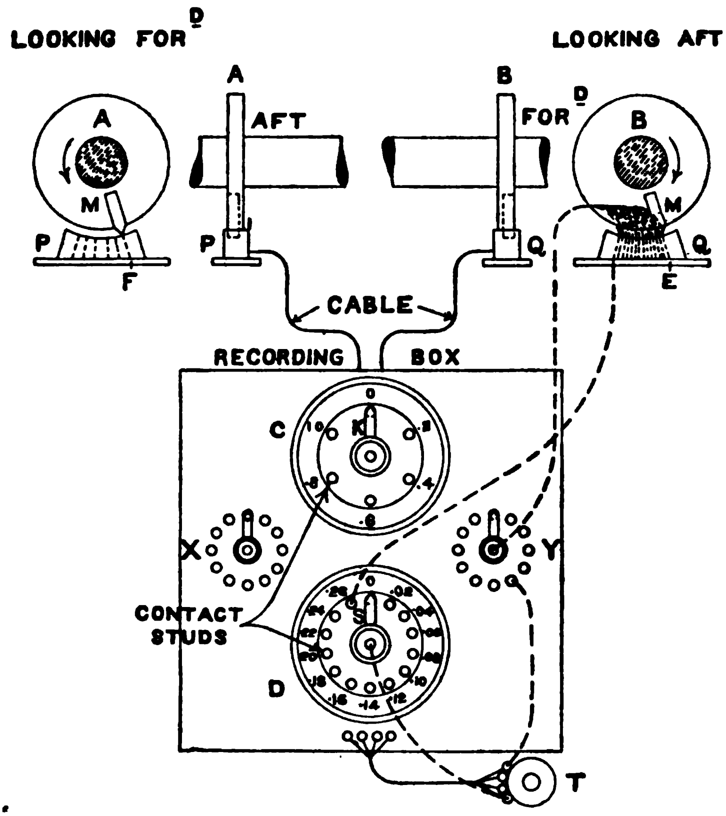


FIG. 339A.

secured to some fixed part of the vessel are the pole pieces of the inductors P and Q, which consist of quadrant-shaped pieces of soft iron, the upper surfaces of which form parts of circular arcs, and are set concentrically with the shaft so as to leave a clearance of about $\frac{1}{16}$th inch between their surfaces and the points of the magnets.

On each inductor, a series of separate but similar coils of insulated wire are wound in a fore-and-aft direction. These coils are spaced uniformly along the length of curved surface; on the forward inductor

Q are fourteen coils spaced about ·02 inches, and on after inductor P are six coils spaced about ·2 inches. In connection with each of these series of coils are recording boxes D and C. These are fitted with revolving contact arms S and K and a series of metal contact studs pitched in a circle and so graduated that the distance between successive studs corresponds with the distances between successive coils of the inductor; thus there are six studs in the recording box C, corresponding to each winding in after inductor P, and fourteen studs in the recording box D corresponding to each winding on forward inductor Q.

The complete circuit for one coil on the inductor Q has been shown diagrammatically in the sketch, and it will be seen that one end of each coil is connected by a separate wire to one particular stud in the corresponding recording box D, the other end of each of the coils on the inductor Q being connected and led by a single wire through the variable resistance box Y to one of the two windings of a differentially wound telephone receiver T and thence back to the contact arm S on the recording box D.

In a similar manner one end of each coil on the inductor P is connected by a separate wire to one particular stud in the corresponding recording box C, the other end of each of the coils on the inductor P being connected and led by a single wire through the variable resistance box X to the other winding of the telephone receiver T and thence back to the contact arm K on the recording box C. The variable resistances X and Y in these two separate circuits enables the strengths of the currents to be initially adjusted.

The wiring of the telephone is arranged so that the currents induced in the coils of the inductors P and Q respectively flow through the telephone windings in opposite directions. The wires between each recording instrument C and D and the corresponding inductor P and Q are efficiently insulated and are enclosed in a single cable.

The forward wheel B is usually set at such a position on the shaft that its magnet is exactly over the first inductor coil in the direction of motion, i.e. E, whilst the after wheel is set so that its magnet is immediately over the last inductor coil in the direction of motion, i.e. F. When the respective contact arms of the recording boxes are placed in contact with the studs corresponding to the coils E and F, and the shaft is revolving but transmitting no power, a current is induced in these coils; this current passes from the coils through the respective contact arms, resistance and telephone windings, back to the coils, and by adjusting the resistances X and Y these currents may be made exactly equal, when no noise will be heard at the telephone receiver since the currents are equal in opposite directions and induced at the same instant.

When, however, the shaft is transmitting power, it is distorted and consequently the forward magnet induces a current in advance of the after magnet, since the angle betweeen the magnet points has been altered by an amount corresponding to the distortion; this unbalanced state will be made manifest at the telephone receiver by a loud ticking noise. If now the contact arm S be shifted from stud to stud, thus successively switching on coils in which the current is induced later than that in coil E, a state of affairs will ultimately be reached when a

balance will be again obtained, either no noise being then heard at the telephone receiver or the sound heard being reduced to a minimum by trial. When this position is found the reading of the contact arm S on the scale of the recording box D will represent the distortion of the shaft at the radius of the inductor windings.

If the reading is too great to be measured by the recording box D, the contact arm K of recording box C is shifted over successive studs until a reading can be taken on the scale of recording box D; in this case the actual reading is the sum of the readings indicated by contact arms of boxes C and D when a balance is obtained.

The horse-power is given by the formula on page 361*c*, but the co-efficient used must be altered to suit the conditions as regards the radius at which the torsion meter magnet is set as compared with that at which the readings were taken when the shaft was calibrated, and the length between the points at which distortion is measured, should the latter be different from that on which the indexes were fitted during calibration.

The later instruments of the Denny and Johnson type are arranged so that the power can also be measured when the shafts are working astern.

Bevis-Gibson flash-light torsion meter.—This consists essentially of two instantaneous shutters arranged to open and close simultaneously, one at each end of the length of shafting under measurement. There are two discs secured to and revolving with the shaft, a fixed lamp, and a torque-finder capable of slight circumferential adjustment (see Fig. 339B). Each disc is perforated near the edge by a small radial slot, and there are similar slots in the lamp mask and at the eye-piece of the torque-finder. The disc slots are in line when there is no twist on the shaft, and at every revolution a flash of light can be seen at the torque-finder. By turning a micrometer spindle at the torque-finder the light is just cut off to right and left respectively, and the readings noted. The mean of the two readings is the 'zero,' which must be deducted from subsequent 'power' readings.

When the shaft twists the shutters fail to synchronise until the torque-finder is moved over the exact amount of the angular displacement of the slots in the discs (see diagrams 1, 2, 3, Fig. 339B). The light is cut off to right and left as before, and the mean reading, minus the 'zero' reading, gives the torque angle in degrees with extreme accuracy.

From observations of the torque angle and of the revolutions, the shaft horse-power may be obtained from the fundamental formula given on page 361*b*, remembering that θ in the formula given thereon is the circular measure of the torque angle.

When the shafting has been calibrated the formula can be used in the form S.H.P. = constant $\times$ N $\times$ a, as explained for the Denny & Johnson instrument.

It is usual in the Bevis-Gibson instrument to measure the distortion in degrees at the vicinity of the rim of the wheel instead of the circumferential linear distortion, as in the Denny & Johnson instrument. The formula for the S.H.P. is therefore in this case S.H.P. $= K_1 \times N \times a$, where a is the torque angle in degrees. The constants K and K_1 in the formulæ for the two instruments are therefore in the ratio of πr to 180.

$$\text{Since } \delta = \frac{\pi r}{180} a,$$

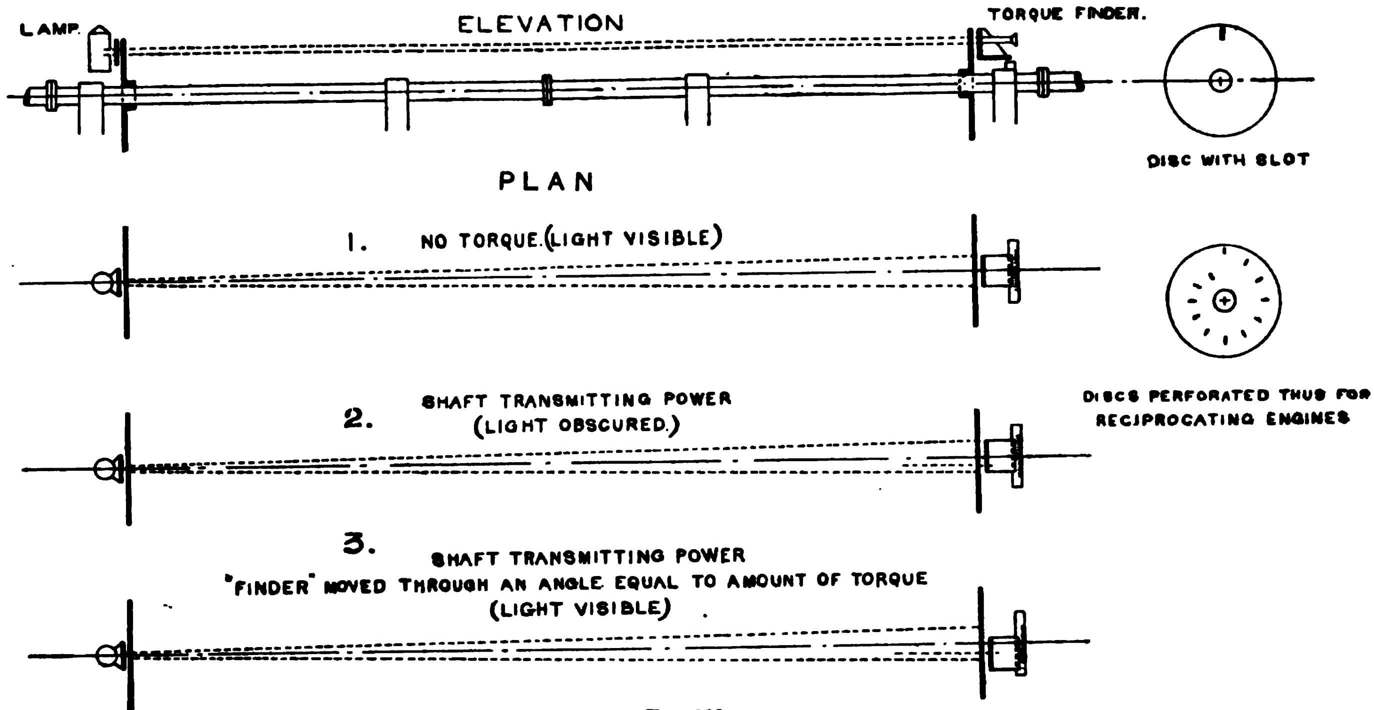

LAMP.
ELEVATION
TORQUE FINDER.
DISC WITH SLOT
PLAN
1. NO TORQUE. (LIGHT VISIBLE)
DISCS PERFORATED THUS FOR RECIPROCATING ENGINES
2. SHAFT TRANSMITTING POWER (LIGHT OBSCURED.)
3. SHAFT TRANSMITTING POWER
"FINDER" MOVED THROUGH AN ANGLE EQUAL TO AMOUNT OF TORQUE
(LIGHT VISIBLE)

FIG. 339B.

r being radius at which the linear circumferential distortion is measured in the case of the Denny & Johnson formula.

It is necessary to have a bright source of light of constant brilliancy, so that the slot in the lamp mask is evenly illuminated. The edges of all the slots should be sharp and clearly defined, and the adjacent parts should be a dull black like the shutter apparatus of an instantaneous camera. Otherwise a scratch at the edge of a slot exposing a bright metallic surface might reflect the light after it is cut off and vitiate the result.

For short lengths of shafting, when the twisting movement available for measurement is relatively very small, the discs are replaced by light concentric drums. The lamp is fixed inside the inner drum next the shaft, and the light ray flashes radially through slots in the drums into a torque finder fitted some distance from the shaft. This arrangement gives the required multiplication of effect without the intervention of gearing.

It will be seen by the foregoing description that the angle of torque is taken at only one point in the revolution, and this is sufficiently accurate for shafting driven by a uniform turning moment such as a steam turbine or an electric motor.

This instrument lends itself readily to the measurement of a fluctuating turning moment such as is produced by a reciprocating engine. In this case the torque angle should be taken at several points in the circle to get a true mean for calculating the shaft horse-power. This is provided for by having twelve slots in each disc, one every 30° at varying radii, as shown in Fig. 339B. The light and the eyepiece of the torque-finder are successively adjusted to the slots at the varying radii, and a complete set of twelve readings is taken when the shaft is running steadily.

General remarks.—The zero reading of the recording instruments of either type should be checked initially by running the engines without any load on the shafting. This is done by working the engines by steam when the shafting aft of the after torsion meter wheel has been disconnected.

The zero can also be subsequently checked underway by *trailing* the shafts—e.g., in a four-shaft arrangement, steam can be shut off one set of engines, and the other set worked at a sufficient speed to cause the shafts of the set from which steam is shut off to revolve by the action of the water flowing past the propellers. In the case of a three-shaft arrangement, also, by working one L.P. shaft by steam under manœuvring conditions the other two shafts can be 'trailed.'

By this means a small reading will be obtained due to the twist necessary to revolve the mechanism forward of the after torsion wheel, and by reference to this reading the accuracy of the zero can afterwards be readily checked. Should a subsequent trailing indicate any material difference in this reading, it would point to the torsion-meter mechanism having become displaced or otherwise defective, or possibly to an abnormal increase in the friction of the turbine mechanism.

Hopkinson-Thring torsion meter.—A third torsion meter, which can be fitted to a small length of shafting, is the Hopkinson-Thring, the principle of which is illustrated diagrammatically in Fig. 339C. A sleeve is clamped on the shaft in the plane C D, the free end having a flange E F,

which is separated by two or three inches from the flange of a collar clamped on the shaft in the plane A B. Bearing surfaces (not shown in the figure) are provided between the collar and the free end of the sleeve which ensure that the relative motion of these two members is one of rotation only, corresponding to the relative motion of the shaft in the planes C D and A B which it is desired to measure. This motion is made visible by means of one or more concave mirrors (two, M, N, are shown in the figure) each of which is carried on the flange of the collar pivoted in a frame in such a way that it can turn about an axis perpendicular to that of the shaft, as at N. Fixed to each mirror is an arm O, from ¼ inch to ¾ inch long, parallel to the shaft axis, a

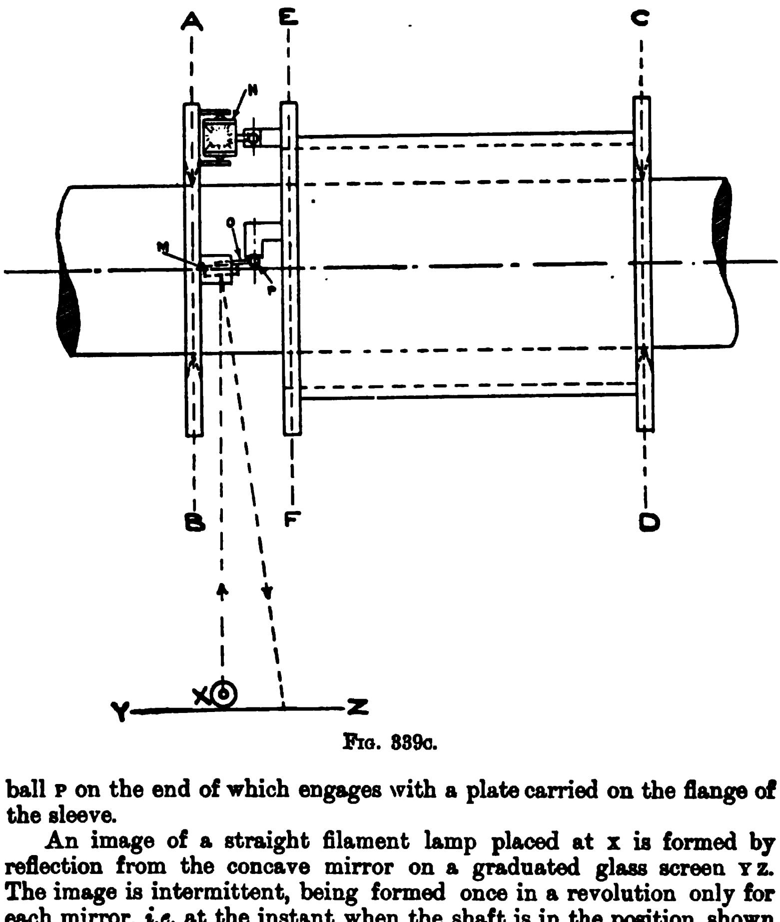


FIG. 339c.

ball P on the end of which engages with a plate carried on the flange of the sleeve.

An image of a straight filament lamp placed at X is formed by reflection from the concave mirror on a graduated glass screen Y Z. The image is intermittent, being formed once in a revolution only for each mirror, *i.e.* at the instant when the shaft is in the position shown in the figure for the mirror M. When the shaft twists the relative movement of the opposed flanges of sleeve and collar causes the mirror to turn about its axis so that the reflection is displaced through an amount proportional to the amount of the twist.

Owing to the small amount of displacement to be measured great

care in design and manufacture is necessary in order to ensure that the relative movement of the sleeve and collar is one of pure rotation and at the same time to avoid introducing too much friction with the constraints necessary for this purpose. The difficulties in securing this have been overcome and the instrument has been fitted in a number of ships.

The zero position of the image on the screen can be obtained by turning the shaft ahead and astern whilst disconnected from the propeller, as described for the Denny-Johnson instrument. The friction of the shaft bearings will cause the reading to be a little different when turning ahead and when turning astern; the mean of the two gives the true zero position.

The displacement of the reflection from the mirror in the Hopkinson-Thring torsion meter gives the torque existing in the shaft at the instant when it flashes across the screen. By mounting several mirrors on the collar flange (such as the two shown in the figure) a corresponding number of reflections is obtained, each of which gives a measurement of torque at a particular point in the revolution. Usually four mirrors are used giving a reading at each quarter revolution. With a turbine the torque is practically uniform and the four readings are the same; they then serve as checks upon one another. When the shaft is driven by a reciprocating engine the torque will vary, but by averaging the readings from the different mirrors the mean torque, and thence the horse-power, can be calculated.

CHAPTER XXVII.

PUMPING, WATERTIGHT, FIRE, DRAIN, AND FLOODING ARRANGEMENTS.

Flow of sea-water into a ship.—The quantity of water that would flow into a ship through a hole may be calculated approximately as follows :—

Let H = depth of hole below the water-line in feet ; A = area of hole in square inches ; and g = accelerating force of gravity.

Then, if V = velocity of flow of feet per second,

$$V = \sqrt{2 g H} = 8\sqrt{H} \text{ approx.}$$

The number of cubic feet of water that would flow into the ship per second is therefore $\dfrac{A \times 8 \times \sqrt{H}}{144} = \dfrac{A \times \sqrt{H}}{18}$.

For example : Supposing a hole, 12 inches in diameter, 16 feet below the surface of the water. The area of this hole is 113 square inches ; so that the rate at which the water would begin to flow into the ship would be $\dfrac{113 \times 4}{18} = 25$ cubic feet per second, or 90,000 cubic feet per hour.

As 35 cubic feet of sea-water weigh 1 ton, a hole 12 inches in diameter, 16 feet below the water, would be capable of admitting into the ship $\dfrac{90,000}{35} = 2,570$ tons of water per hour. It is evident, therefore, that a comparatively small hole, especially if at any considerable depth below the water-line, would require a very much larger pumping installation to cope with the influx of water than is provided even in the largest warship (see Table I.).

In cases where injury is so serious as to admit large bodies of water, the pumping appliances would not be powerful enough to keep the compartment free, unless collision-mats, or other leak-stoppers, could be effectively applied to the damaged part. The only complete safeguard is the division of the hull into watertight compartments, so that the effects of any injury may be localised, and in cases where the ship sustains considerable damage below the water-line, such as from collision, &c., the steam pumping appliances must only be regarded as an auxiliary. The pumps are, however, very valuable for dealing with the ensuing leakage in adjacent compartments due to local straining of bulkheads, double-bottoms, &c.

Watertight compartments.—These are formed by steel watertight bulkheads built across the ship at certain sections. In warships there is generally further sub-division by a longitudinal bulkhead at the middle line of the vessel ; also by bulkheads in the coal bunker and wing spaces ; and by horizontal steel watertight decks. In many warships the middle-line bulkhead extends throughout the engine and

boiler rooms, but in the more modern it is confined to the engine room only. In some recent ships, with a view of still better maintaining the watertight subdivision and of limiting the amount of the heel in case of accident, the engine rooms are divided into three compartments by longitudinal bulkheads and in some cases are further subdivided into five watertight compartments by transverse bulkheads. This middle-line bulkhead has also become general in passenger steamers which are fitted with twin screws. When the principal vertical bulkheads do not terminate at a watertight deck, their upper edges should be carried to a sufficient height above the water-line to prevent the water in an injured compartment flowing over the tops of the bulkheads into the adjoining compartments, even when the ship is at the greater immersion due to the compartments being full of water. The height to which these bulkheads are carried above the water-line should be proportioned to the volumes of the respective compartments, in order to ensure safety without unduly increasing weight.

Double-bottom.—The safety of most ships in the Royal Navy is still further increased by the construction of a 'double-bottom.' This consists of an inner watertight skin, at some distance from the outer skin, extending for about two-thirds to three-fourths of the total length of the ship. The distance between the two skins is generally about 3 or 4 feet, depending on the size of the ship ; above the turn of the bilge, the inner skin is continued by vertical bulkheads carried up above the water-line, and forms, with the outer skin, what are called the 'wing' compartments. The double-bottom and wing compartments are subdivided into many small compartments by longitudinal and transverse bulkheads. In H.M.S. 'King Edward VII.' there are 106 compartments in the hold space, 60 in the double-bottom, and 38 in coal bunkers and wings, making a total of 204 watertight compartments.

Considerable damage may be done to the outer skin of a ship with a double-bottom without endangering her safety, for unless the inner skin be broken no water can enter the hold. The compartments in the double-bottom are so small, that the filling of several would have comparatively little effect on the immersion or the trim of the ship.

Reserve feed-water and oil-fuel tanks.—Some of the double-bottom compartments, generally under the boiler rooms, have been utilised as stowage tanks for reserve fresh water for the boilers. In recent vessels some double-bottom compartments not used for the stowage of fresh water, are utilised for carrying oil fuel. Until recently these tanks were connected by sluice valves ; but in the latest ships this practice has been discontinued, each tank being generally fitted with its own filling and emptying pipes.

Access to the double-bottom and wing compartments, enabling the plating to be examined, cleaned, and painted, is provided by manholes in the inner skin, and two of these manholes are fitted to each of the compartments at opposite ends to facilitate ventilation.

When the compartments are not under examination or repair these manholes are closed by plate covers, which, in the case of the ordinary compartments, are secured by nuts and bolts, and the joint is made with red-lead putty. In fresh-water compartments india-rubber is substituted for red lead as the jointing material, and in oil-fuel compartments the joint is made with liquored leather, and the cover

is secured by a type of nut which can only be worked by a special spanner which is accessible only to authorised persons. The use of hinged covers for closing manholes of double-bottom and wing compartments, jointed by the edge of an angle-bar on india-rubber with butterfly nuts, is being discontinued.

Sounding tubes with screwed caps are fitted to all compartments used for stowage of fresh water or oil fuel, and in the case of the oil-fuel tanks the caps are provided with locking arrangements. The reserve fresh-water tanks are each provided with an air-escape pipe, carried well above the water line, and each oil-fuel tank is fitted with two air-pipes led to the upper deck from the opposite ends of the highest parts of the tanks.

To allow for the escape of air from ordinary double-bottoms of war vessels, should it be found necessary to flood them, a 1½-inch diameter screwed plug is fitted to each manhole cover (see Fig. 340). To prevent any possibility of water entering the bilges when the compartment is full through the plug being mislaid, it is arranged so that

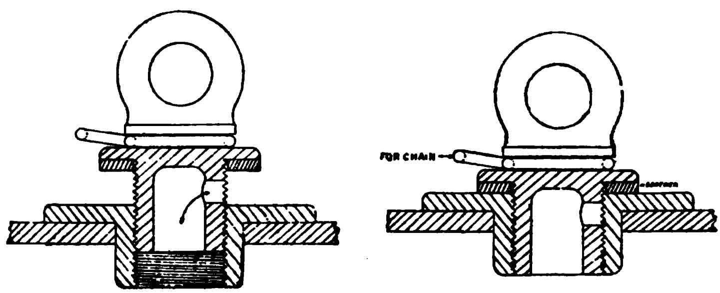
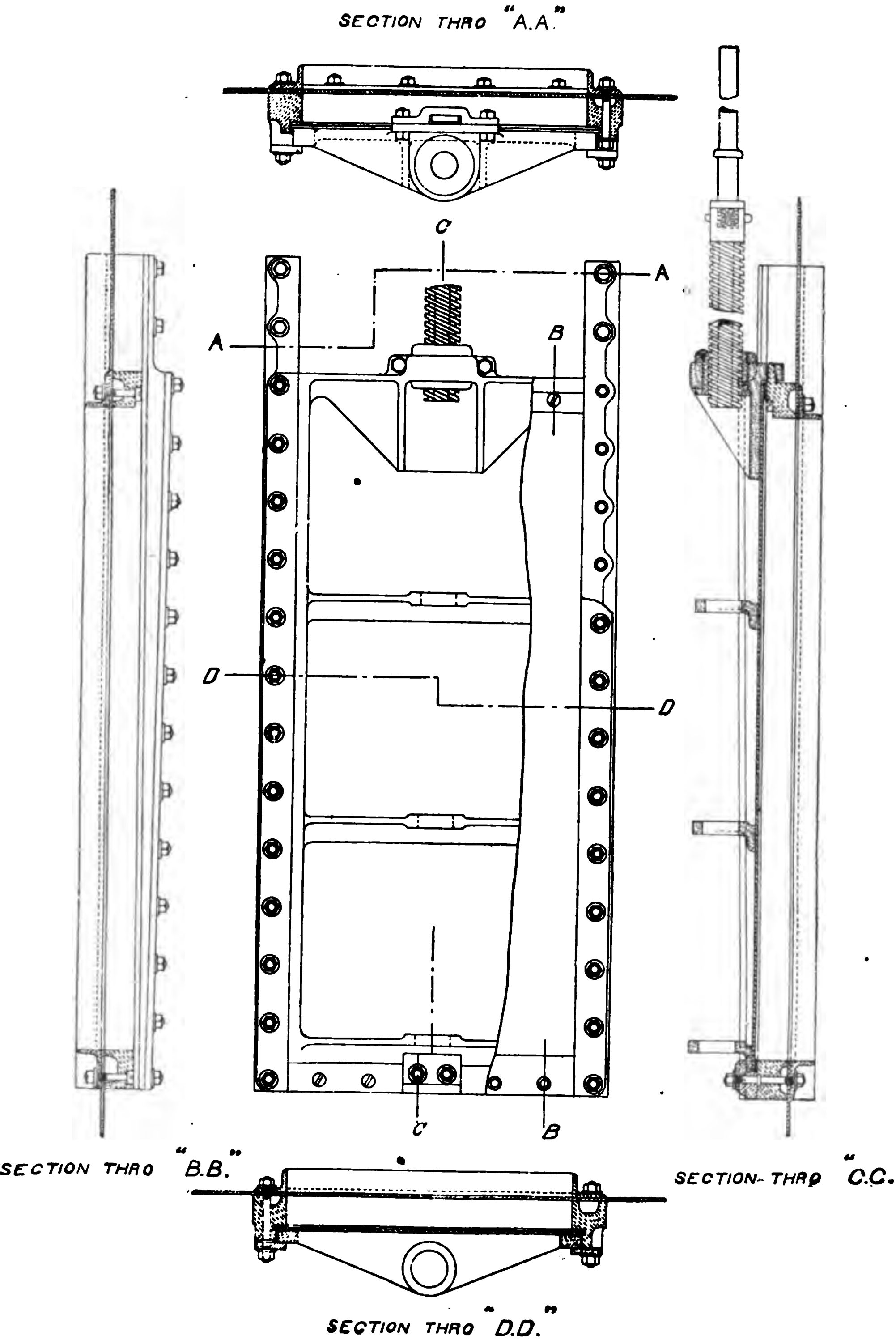


FIG. 340.

when unscrewed two or three turns of the thread, air can escape from the compartment through a small hole as shown on the left of Fig. 340. When water emerges the plug is screwed down, as shown on the right of the figure. In the mercantile marine the double-bottom tanks are used for water-ballast purposes, water from the sea being admitted as may be required.

Watertight doors.—The present practice is to eliminate watertight doors and hatches as far as possible, there being no communication below the water-line between the various principal compartments. In vessels where communication is maintained, sliding watertight doors are fitted (see Figs. 341, 342 and 342A).

When the door is worked in a vertical direction it is raised and lowered by means of a screw (see Figs. 341 and 342A). When the watertight door slides horizontally, it is worked by means of racks and pinions, the racks being fixed on the back of the door, as shown in Fig. 342.


SECTION THRO "A.A."
A
A
B
C
D
D
C
B
SECTION THRO "B.B."
SECTION THRO "C.C."
SECTION THRO "D.D."
FIG. 311.

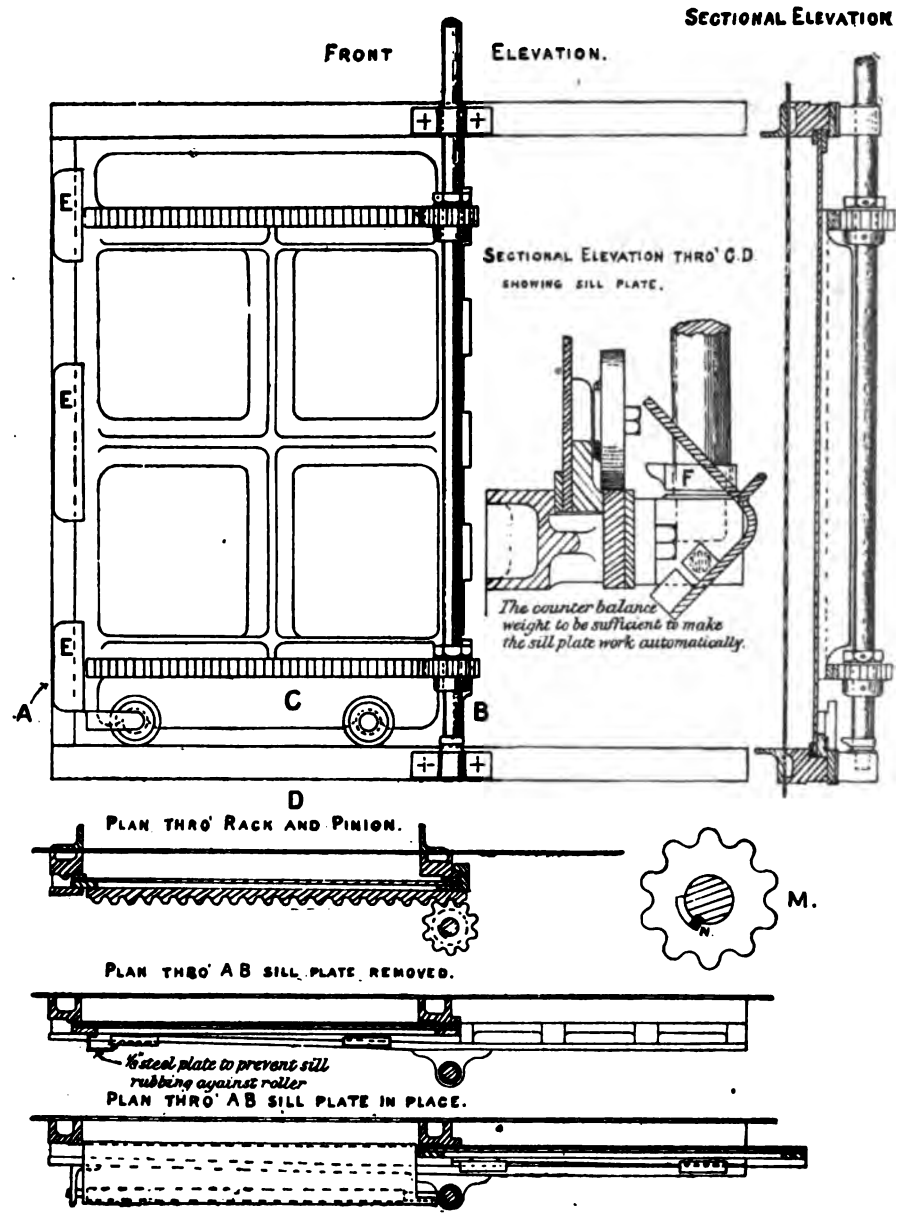

SECTIONAL ELEVATION
FRONT ELEVATION.
SECTIONAL ELEVATION THRO' C.D
SHOWING SILL PLATE.
F
The counter balance weight to be sufficient to make the sill plate work automatically.
E
E
E
A
C
B
D
PLAN THRO' RACK AND PINION.
M.
N.
PLAN THRO' A B SILL PLATE REMOVED.
⅛" steel plate to prevent sill rubbing against roller
PLAN THRO' A B SILL PLATE IN PLACE.

FIG. 342.

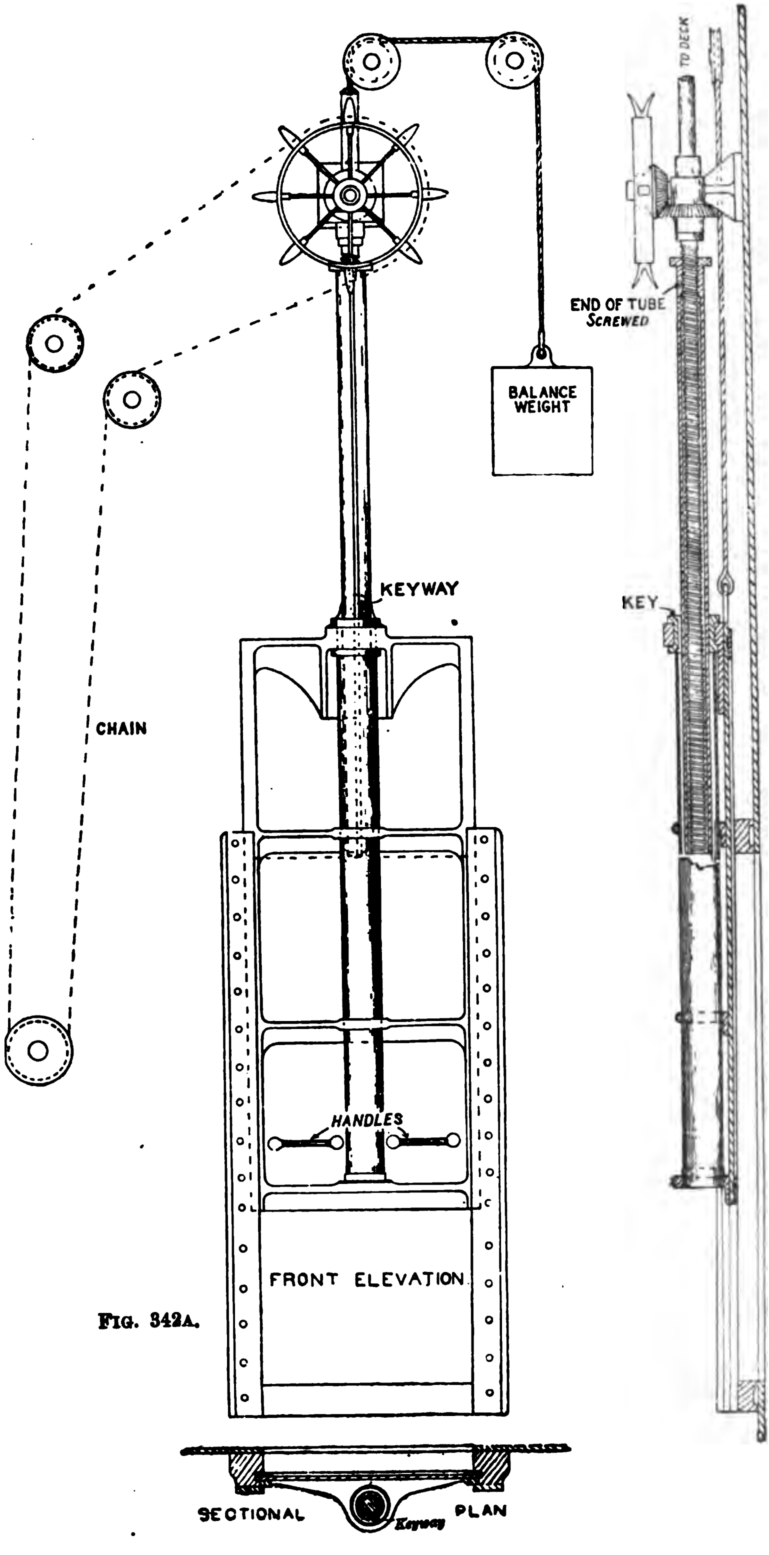

TO DECK
BALANCE WEIGHT
END OF TUBE SCREWED
KEYWAY
KEY
CHAIN
HANDLES
FRONT ELEVATION
SECTIONAL PLAN
Keyway

FIG. 342A.

The spindles for working these doors are led to the main or upper deck, above the water line, and are placed in positions readily accessible in case of emergency; means are also provided for working them at or near the doors. Wedges E (Fig. 342) are fitted on the door and at the ends of the guides, so that when the door is shut it is pressed tightly against the inner face of the guide to prevent the passage of water. To promote uniformity and prevent mistakes, the gear for working water-tight doors is fitted so that the door is closed by a right-hand motion. Every precaution should be taken to keep the guide grooves clean and the gear in good working order, so that there may be no difficulty in closing the doors in case of necessity, when any mistake may not only be inconvenient, but fatal.

In the case of horizontal doors the bottom grooves are very liable to get choked, and to prevent this a plate or sill is arranged to cover the groove when the door is open. The sill is worked automatically by the spindle feather N being allowed play in the pinions as shown at M, so that before the pinion can communicate any motion to the rack towards closing the door, the cam F keyed on the spindle engages with the stop on the sill and turns it about the fulcrum clear of the path of the door. When the door is opened wide the sill is clear of the door and closes by means of the counterbalance weight. Where doors are fitted in the hold space, e.g. the communication doors between the engine and boiler rooms, they are kept well above the inner bottom and special provision is made for quickly closing them.

Vertical doors are capable of more rapid and certain closing than horizontal doors, and as height is generally available, are fitted for all important doors in the hold space. The quick closing is generally effected by a quadruple-threaded screw of coarse pitch engaging with a nut on the door, while for ordinary vertical closing doors the screw is a double-threaded one. Fig. 342A shows a satisfactory method of working a vertical quick-closing door, which is fitted in many warships. The door is balanced and can immediately be pulled shut at the door itself. It will be seen that it can only be opened from the below position, and the screw is only used for closing the door from the upper position or for jambing it tight at the lower position after it is pulled shut. When the screw has been used for closing the door, it must be unscrewed before the door can be lifted.

On the decks above the water line and for minor watertight compartments hinged doors are fitted (see Figs. 342B and 342C). They are made watertight by indiarubber strips secured to the doors, which by the action of clips on wedge pieces fixed to the doors, are pressed tightly against the edge of the angle-iron forming part of the structure of the door frame. The clips, which are spaced at about equal distances around the edge, are workable from both sides of the door, and springs are fitted where necessary to support the clip handles when the door is open. Modifications of this type of door are fitted for entrance to magazines, escapes from bunkers, access to wings, &c.

Electric hoists.—In recent ships, owing to the omission of doors in the bulkheads of the main machinery compartments, electric hoists have been fitted in order to facilitate communication between the various machinery compartments. In addition they are designed for

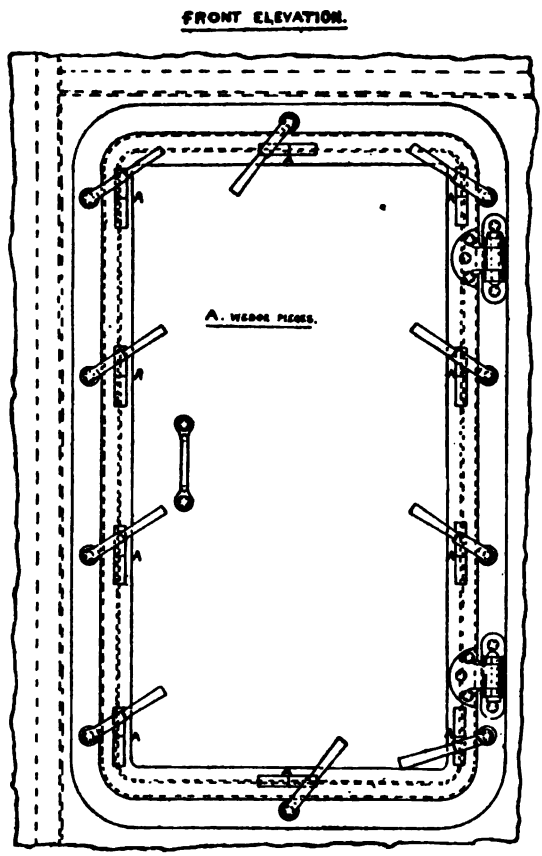


FIG. 342B.

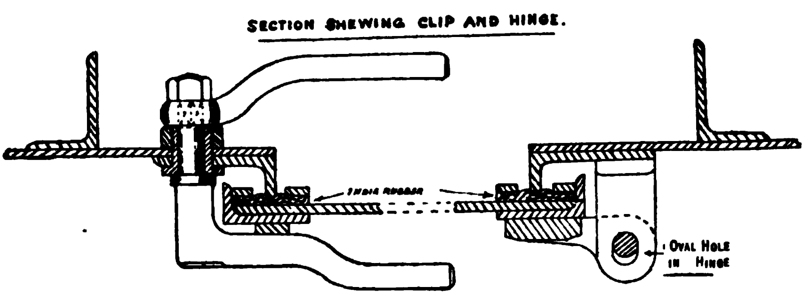


FIG. 342C.

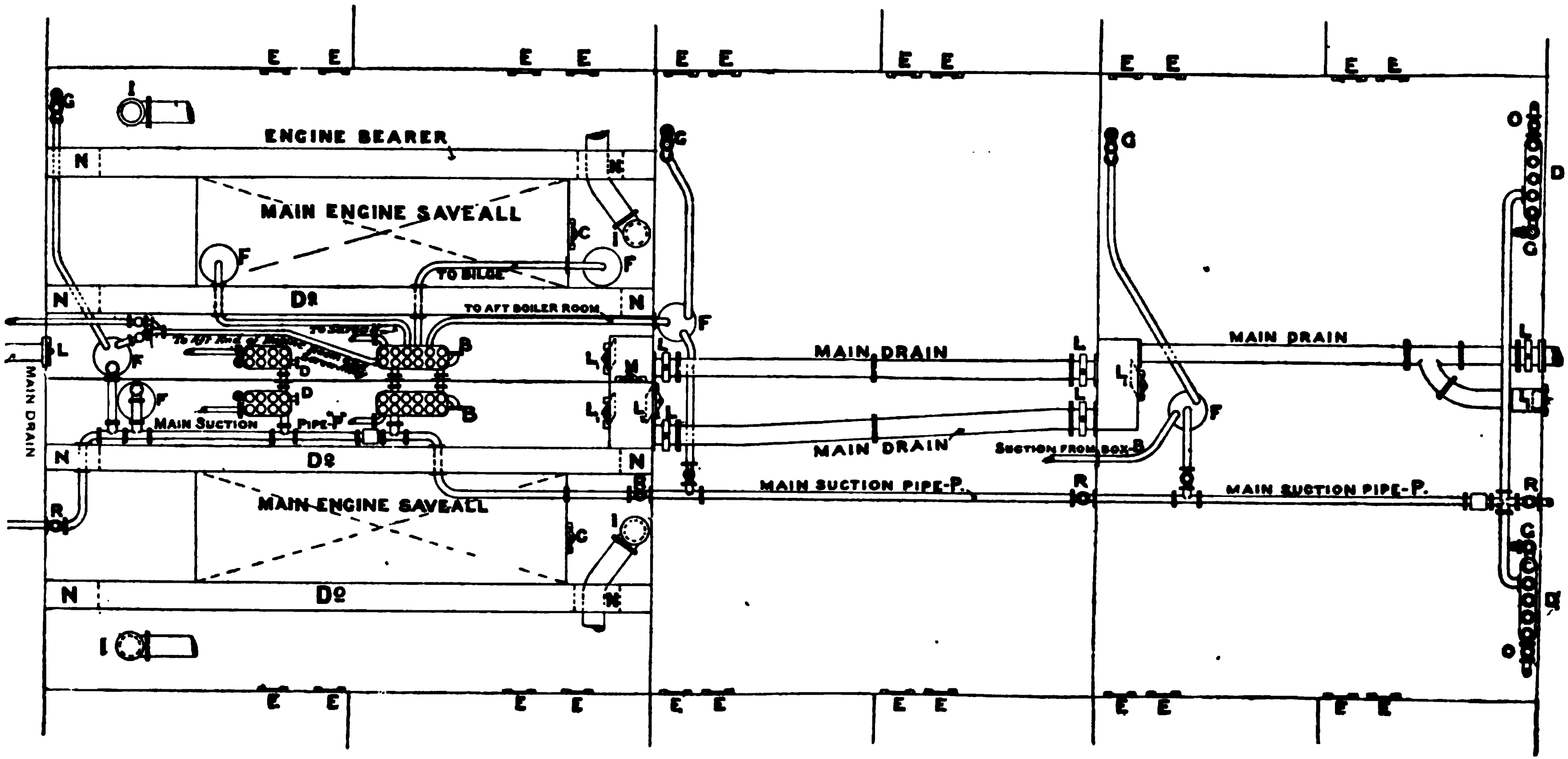

ENGINE BEARER
MAIN ENGINE SAVEALL
TO BILGE
TO AFT BOILER ROOM
MAIN SUCTION
PIPE-P
MAIN DRAIN
MAIN DRAIN
MAIN DRAIN
SECTION FROM BOX-B
MAIN SUCTION PIPE-P.
MAIN SUCTION PIPE-P.
MAIN ENGINE SAVEALL
MAIN DRAIN

FIG. 342D.

the ready removal of wounded and are utilised for the removal of ashes, dirt, &c.

Sluice-valves.—Excepting in most recent vessels in which the bulkheads are kept practically intact, small sluice valves are fitted in the lower parts of the watertight bulkheads (Figs. 342D and 342E) to allow the water to be drained from one compartment to another, or to afford communication for levelling purposes between compartments, such as adjacent reserve feed water or adjacent oil-fuel tanks. These valves are arranged to shut with a right-hand motion, and the rods, &c., for working them are carried to the same positions as the gear for working the watertight doors. They are also fitted so that they can be opened and shut locally.

Pumping arrangements.—The pumping arrangements should be of such a nature that in addition to being available when the main pro-

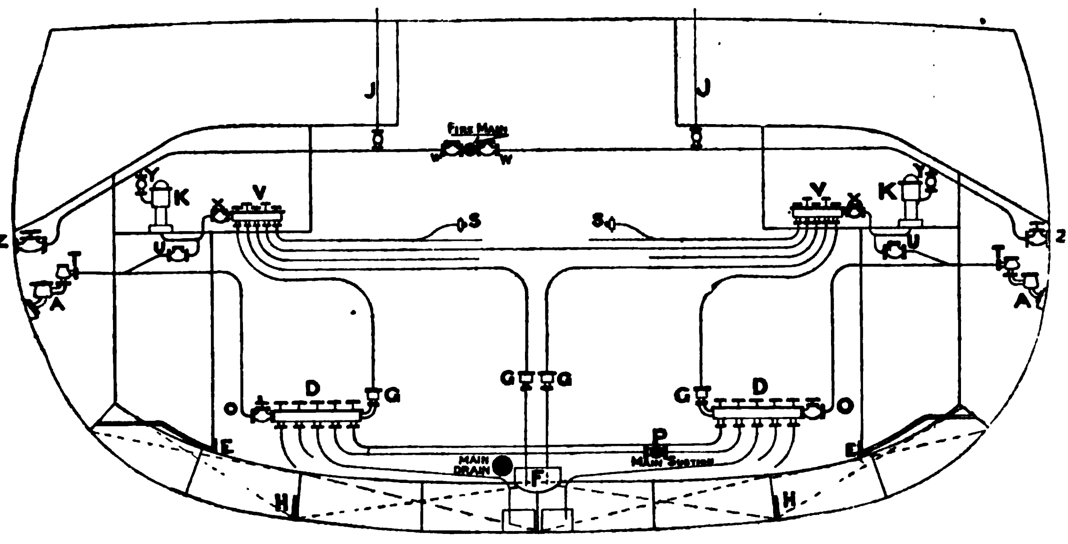


FIG. 342E.

pelling engines are not in use they can also be used whilst the main engines of the ship are at work, so that in case of accident the ship may be able either to proceed to the nearest port, or in an extreme case, that the ship may be kept afloat long enough to be run into shallow water or be conveniently beached.

The usual fittings are :—(1) Each main circulating pump is fitted with a bilge suction pipe, and pipes are provided so that large quantities of water can be drained to the engine room bilges from the principal compartments of the vessel and be dealt with by these suctions. (2) Special pumps worked directly off the main engines and fitted with suction pipes so that they can be connected with all the principal ship compartments. They are now rarely fitted in warships. (3) Separate bilge pumps. These have suction as in (2) and in addition are fitted so that they can be used as fire pumps. (4) Steam

TABLE I.

—	Circulating Pumps		Pumps worked direct from Main Engines (Save-all Pumps)		Separate Bilge Pumps		Steam Ejectors		Hand Pumps		Total.
	No.	Capacity. Tons per hour each	No.	Capacity. Tons per hour each	No.	Capacity. Tons per hour each	No.	Capacity. Tons per hour each	No. and Size	Capacity. Tons per hour each	Tons per hour, neglecting Save-all and Hand Pumps
Battleships and 1st Class Cruisers	4	1200 to 1500	2 to 4	8 to 12	4	80 to 100	Nil	—	4 of 9″ dia.	25	5120 to 6400
2nd Class Cruisers . .	2	850 to 1200	2	6 to 8	4	60 to 75	Nil	—	4 { 2 of 9″ dia. 2 of 7″ dia.	25 13	1940 to 2700
3rd Class Cruisers . .	2	600 to 850	2	4 to 6	2	40 to 80	Nil	—	3 { 1 of 7″ dia. 2 of 6″ dia.	13 10	1280 to 1860
Scouts	2	850	2	3 to 5	2	80	Nil	—	3 { 1 of 7″ dia. 2 of 6″ dia.	13 10	1860
Torpedo Boat Destroyers .	2	200	Nil	—	1	10 to 15	6	40	1 of 5″ dia.	6	650 to 655

TABLE II.

—	Circulating Pumps		Separate Bilge Pumps				Total
			Steam		Electric		Tons per hour
	No.	Capacity in tons each Engine-room per hour	No.	Capacity in tons per hour each	No.	Capacity in tons per hour each	
Battleships	4	1400	7 { 4 3	75 50	7 { 5 2	50 30	3560
1st Class Cruisers	4	2000	8 { 4 4	75 50	10 { 2 8	50 30	4840

ejectors. These are used generally only in connection with the compartment in which they are fitted, and in ships where the weight and space for ordinary bilge pumps of adequate capacity cannot be allowed. (5) Hand pumps which are connected with a general system for dealing with all compartments in case steam is not raised.

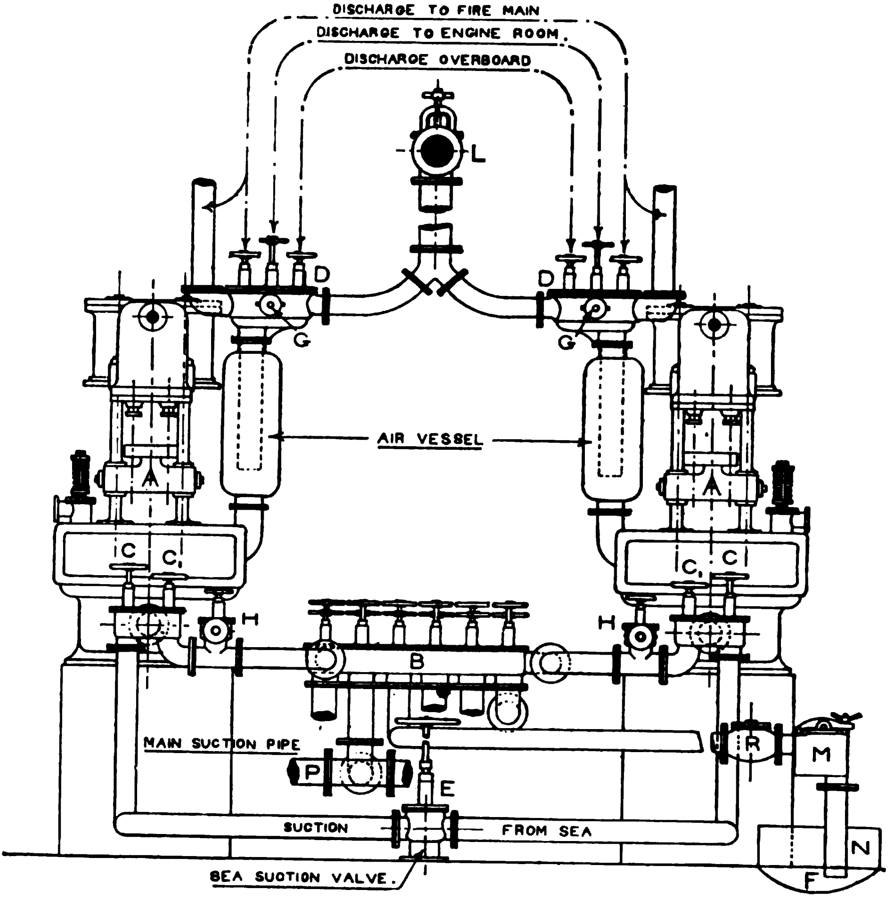
With the watertight subdivision which has hitherto obtained, the pumping capacity available in warships of the Royal Navy is shown in Table I.

In recent warships where the number of watertight doors has been limited, each main transverse compartment is fitted with its own pumping installation, at least one pump being fitted in each compartment. Where steam is readily available—as in the engine and boiler rooms—the pumps are steam-driven; in other cases they are driven by electric motors. Two steam pumps are fitted in each engine room, so that one would always be available for fire or deck purposes. In vessels where oil-driven dynamos are fitted, which ensure electric power being available, even though steam is not raised, the use of hand pumps has been practically discontinued.

In these recent ships the drainage from outside compartments is not led to the engine rooms; hence the bilge suctions from the main circulating pumps have been omitted or only one main circulating pump suction of limited diameter has been fitted in each engine room, and in the latter case, though the pumping capacity of each pump remains the same, the single suction pipe limits the total pumping capacity of these pumps to about half that formerly provided. The pumping power is shown in Table II.

The pipe arrangements provided in the older larger warships referred to in Table I. are shown in Figs. 342D and 342E. The principal fittings for dealing with large bodies of water which may enter the ship through the effect of collision or similar accidents, are the bilge suctions I of the main circulating pumps. In case of water entering the engine rooms, these pumps can deal with it direct, and from compartments outside the engine rooms the leakage is led to the engine-room bilges through large galvanised steel pipes called the main drain system.

Main Drain System.—The system extends throughout the length of the double-bottom, with branches carried forward and aft to drain compartments on and above the platform deck. The pipes are about 15 inches in diameter or of equivalent clear area through the boiler rooms, and where they deliver into the engine rooms, but before and abaft these spaces are gradually decreased in diameter. They are fitted in duplicate in the two after boiler rooms, sluice valves being arranged so that each run can be used independently. The pipe is omitted in the engine rooms, and the bilges of these compartments are regarded as tanks into which the drainage from forward and aft can be led. Large sluice valves, L L, are fitted to the main drain at each bulkhead, and at the branches from the principal compartments to regulate the flow of water. The engine-room bilges are cross-connected by means of a sluice valve M, at the middle line bulkhead, and the engine bearers, &c., are arranged with large water courses, N, to conduct the water to the circulating pump bilge suctions I. The sluice valves are capable of being worked from the main deck and from the hold in the vicinity of the valve. Each sluice valve opening (L_1) to the main


DISCHARGE TO FIRE MAIN
DISCHARGE TO ENGINE ROOM
DISCHARGE OVERBOARD
L
D
G
AIR VESSEL
A
C
C,
H
B
MAIN SUCTION PIPE
P
E
R
M
SUCTION
FROM SEA
SEA SUCTION VALVE.
N
F

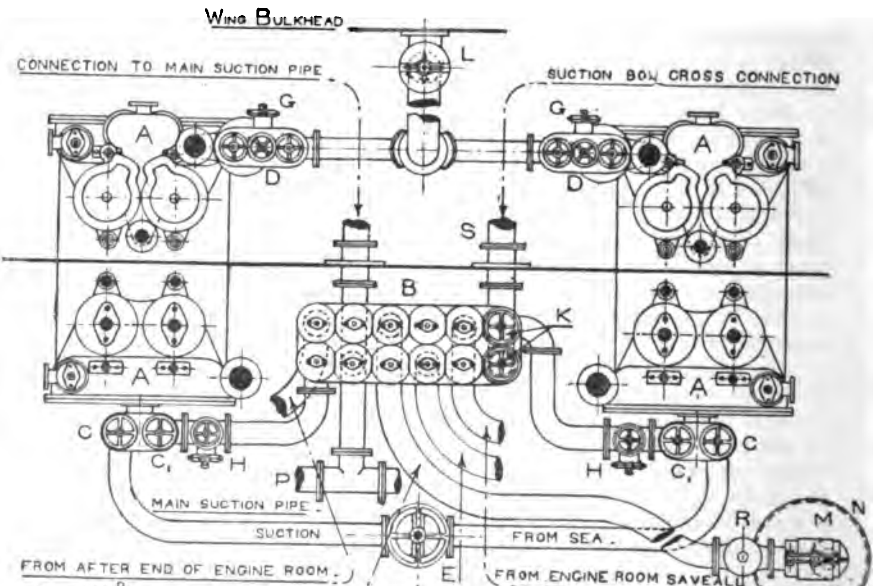

WING BULKHEAD
L
CONNECTION TO MAIN SUCTION PIPE
SUCTION BOX CROSS CONNECTION
G
A
D
S
B
K
C
C,
H
P
MAIN SUCTION PIPE
SUCTION
FROM SEA
R
M
N
F
E
FROM AFTER END OF ENGINE ROOM
FOR^D
FROM ENGINE ROOM SAVEALL
BOILER ROOM

FIG. 342F.

compartments is protected by a non-return valve as shown, to prevent water from one compartment flowing back into another if at any time the sluice valve is unintentionally left open; strainers are arranged in connection with all openings on the side from which the water is drained.

From wing compartments and bunkers abreast the main machinery spaces water can be drained by sluice valves, E E, into the adjacent bilges. The ammunition passages are also drained to these bilges, a lift and a non-return valve being fitted on each pipe. Drainage from the compartments on and above the double-bottom beyond the machinery spaces, including the wings, is generally led either direct or by branch pipes to the main drain, non-return as well as sluice valves being generally fitted on any connecting pipes. Shell-rooms, magazines, spirit-rooms, which are generally situated in these spaces, are not always fitted with special drainage arrangements. These rooms are liable to little or no drainage, and are seldom flooded. In the latter case the water is generally dealt with as on page 371*f*. In some cases the connections from the wing compartments are without non-return valves. Drainage from compartments on and above the platform deck before and abaft the double-bottom is conducted to the main drain by branches, each branch being fitted with a screw-down and a non-return valve in the compartment. The adjacent wings are drained by sluice valves to these compartments.

The main drain also serves the purpose of draining to the engine room bilges any water which may enter the ship under working conditions, e.g. from submerged torpedo tubes when running out bars; from barbettes when washing out guns; the circulating water of air-compressor cooling jackets when this is discharged inboard; from cable lockers and similar spaces where water is likely to accumulate. This drainage can be dealt with by the separate steam bilge pumps. Arrangements are made for flushing the main drain pipes from the ends to the engine room bilges direct from the sea, and chains are sometimes fitted for clearing the drain pipes when choked.

For convenience the main drain pipe was, in the older ships, carried through, but not in connection with, the double bottom, and was fitted only for the drainage of compartments in the hold. Various accidents to the ship's outer skin and consequent fracture of the drain pipe having shown the danger of such a lead, they are now placed above the inner bottom. In many such vessels the main drains are connected to large drain cisterns, and the separate steam and hand pump suctions are led to these cisterns, which overflow into the engine room bilges when the circulating pumps are required. In later ships where watertight doors are not fitted in the bulkheads below water line the main drain pipe has been omitted.

Suction and discharge arrangements of the separate fire and bilge pumps.—For smaller amounts of leakage the fire and bilge pumps have suctions leading direct to the forward and aft ends of engine rooms, with a continuation to the screw tunnel from the latter; to the main engine save-all, to each boiler compartment, and to the main suction pipe described hereafter. They are also fitted with suctions to the sea (Fig. 342F). The pumps can deliver either overboard direct, to the engine-room through a valve with a hose connection, and to the fire main, a large air vessel being fitted in connection with

the discharges. Fig. 342F shows a general arrangement of the suction and delivery pipes of a large ship, the pumps in each engine

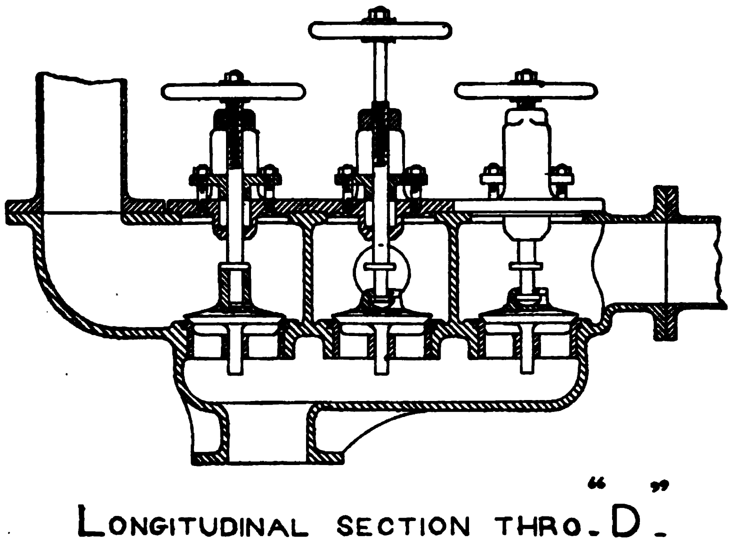


FIG. 342G.

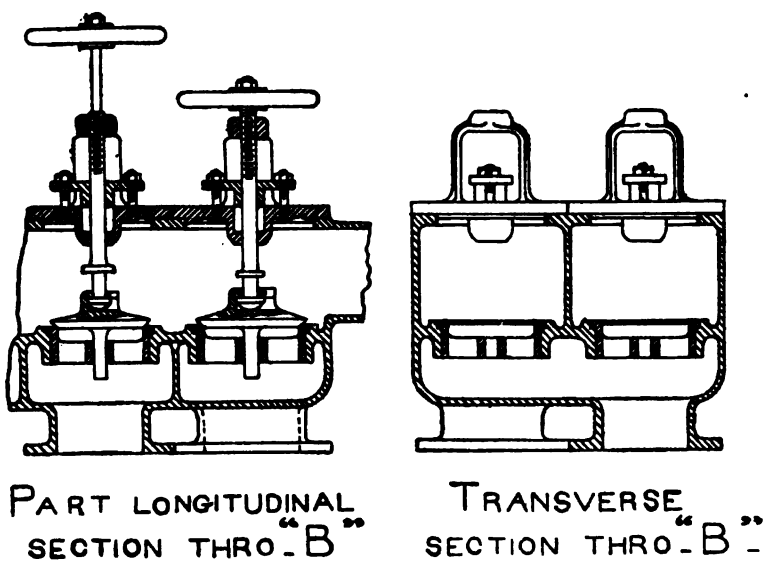


FIG. 342H.

room having similar suction and delivery arrangements. A A are pumps. B B are directing valve boxes to which the bilge suctions are

led, O O_1 are shut-off valves from the sea inlet valve E and the bilge directing valve-box B respectively, so that one pump in each engine room can pump from the sea whilst the other pumps from the bilge. Valves H with hose connections are provided on the suction pipes, so that hoses can be led to any part of engine room, in case the ordinary suction should be choked. The directing suction valve boxes B B are cross-connected between the engine rooms by a pipe S, and by lift valves K K in these boxes, so that each pump can be used on the suctions in each engine room. The shut-off valve O_1 from the bilge directing box is of the screw-down non-return type to prevent accidental flooding of the bilges in case the shut-off valve O from the sea is left open; in some cases the valves in the directing valve-box B B, excepting K K, are also of the screw-down non-return type for this purpose. D D are directing discharge valve boxes by means of which the water may be pumped overboard through sea valve L, to the engine room through a branch with hose connections G, and to the fire main, the latter having a non-return valve. The sea valve L is a non-return valve, which can be lifted by hand but cannot be jambed shut. The valve seats in the suction and discharge directing valve boxes B B and D D are made separate from the main castings for purposes of ready renewal as shown in the sections Figs. 342G and 342H.

The connection to the main suction P permits the steam pumps to be used on all the double bottoms and bilges, as explained later. Should the main engine save-all suction of these pumps or of the save-all pumps themselves become choked, the sluice valve O (Fig. 342D) affords a direct connection with the engine-room bilge suction, and in case of necessity the hose connections at H (Fig. 342F) can be used.

Main suction pipe.—The separate fire and bilge pumps are connected through valves on the directing valve boxes B B with a pipe termed the main suction, P P P (Figs. 342D, E, and F), which in large ships is 6 inches in diameter in wake of the double bottom, and tapers slightly from this part to the ends of the vessel. This pipe has branches leading to the lower valve boxes D (Figs. 342D and E), and thence to double bottoms; to bilges forward and aft of double bottom; and to pockets F in the bilges of the machinery spaces. In order to avoid accidentally emptying the double bottoms used for the storage of reserve feed water or oil fuel, or to prevent the accidental flooding of these spaces by sea water, the suction pipes to these spaces are broken below the lower valve boxes D, and hose connections are fitted on the valve boxes, and on the ends of the suction pipes, special hoses being supplied for connecting when necessary. Stop valves R, workable from the middle or main deck, are fitted on the main suction pipe at every bulkhead through which it passes, and the branch suction valves are also worked from similar positions.

Hand pump connections.—In warships referred to in Table I., hand pumps of numbers and sizes stated therein are fitted; and the general arrangement of these pumps K, with their connections, is shown in Figs. 342D and E. They have suctions leading to the lower valve boxes D, and through them to all the double-bottom compartments and main suction pipe; to the bilges forward and aft of double bottoms; to the main machinery compartment bilges; to the magazines, shell and spirit rooms by means of portable hose connections to

valves S S; and to the sea through valves A, T and U. The suctions are brought to the common directing valve-box V V situated near the pump, and the pump suction is taken from this box through a non-return valve X. Non-return valves G are fitted on all vertical suctions. As in the case of the separate bilge pumps, it will be seen that the hand pumps can deal with water in any part of the ship. The discharge is provided with a non-return valve Y at the pump, and can be directed overboard through sea valve Z; to fire main through a screw-down non-return valve W; to auxiliary rising mains J on various decks. The latter are independent of the fire main and are provided to supply water for fire, wash-deck purposes, etc., when steam is not available.

General.—We thus see that the usual drainage and a leakage of moderate amount can be dealt with as follows (Figs. 342D and E).

(1) In double bottoms:—By the separate steam bilge pumps through the main suction pipe, or by the hand pumps through the lower valve boxes. (2) Machinery spaces over double bottoms:—By the separate steam bilge pumps either direct through their separate suctions, or through the main suction pipe; by the hand pumps, either direct or through the main suction pipe. (3) Other spaces over double bottoms:—Some of the compartments can be drained into the main drain; others, such as shell rooms, magazines, and spirit rooms, are more generally emptied by portable hoses as referred to above. (4) From bunker compartments abreast machinery spaces:—By draining to machinery compartment bilges through sluice valves at the bulkhead and then as in (2). In later ships the sluice valves have been omitted, the watertight door openings being used for this purpose. (5) From wing compartments abreast machinery spaces, which compartments, especially abreast the boiler rooms, are now nearly always used as bunkers:—by draining to machinery compartment bilges through pipes with sluice valves at the bulkhead, and then as in (2). (6) From wing compartments beyond the machinery spaces:—by draining direct to the main drain. (7) Compartments on and above platform deck forward and aft of double bottoms:—by draining direct to main drain. (8) Bilges before and abaft double bottom:—by the separate steam bilge pumps or hand pumps through main suction pipe, or by hand pumps direct through their separate suctions. (9) Sediment in oil fuel tanks:—by a portable hand pump which is connected by a hose to the double bottom suction pipe (see page 371f), and the sediment is discharged overboard through hoses.

Later modifications.—In the later ships in Table I, the pumping and draining arrangements have been simplified as follows:

(a) *Main drain.*—The pipes are not fitted in duplicate in after boiler room, and the pipe in forward boiler room has been omitted, sluice valves being arranged in the forward bulkhead of this compartment, as in the case of engine room.

(b) *Separate fire and bilge pumps.*—Instead of having separate suction pipes leading to each boiler room, a single pipe is carried forward, having branches with a screw-down non-return valve to each boiler room bilge pocket.

(c) *Main suction.*—The main suction pipe has branches leading direct to all double bottom compartments, the lower valve boxes D are not fitted, and the hand pumps are connected direct to the main

suction pipe. At the junction of each suction branch for the double bottoms with the main suction a special valve is fitted (Fig. 342J). When the cotter is in place the valve can only be used as a screw-down non-return. When the padlock is unlocked the cotter can be removed and the spindle can be raised so as to make it a lift valve, and it can then be utilised for flooding purposes. The valves on the branches to double bottoms are arranged to be worked from suitably high positions in the hold. In some cases the branches from the main suction pipe to the machinery compartment bilges have been omitted.

(*d*) *Sediment in oil fuel tanks.*—A separate discharge valve is fitted in each main machinery compartment, so that this sediment can

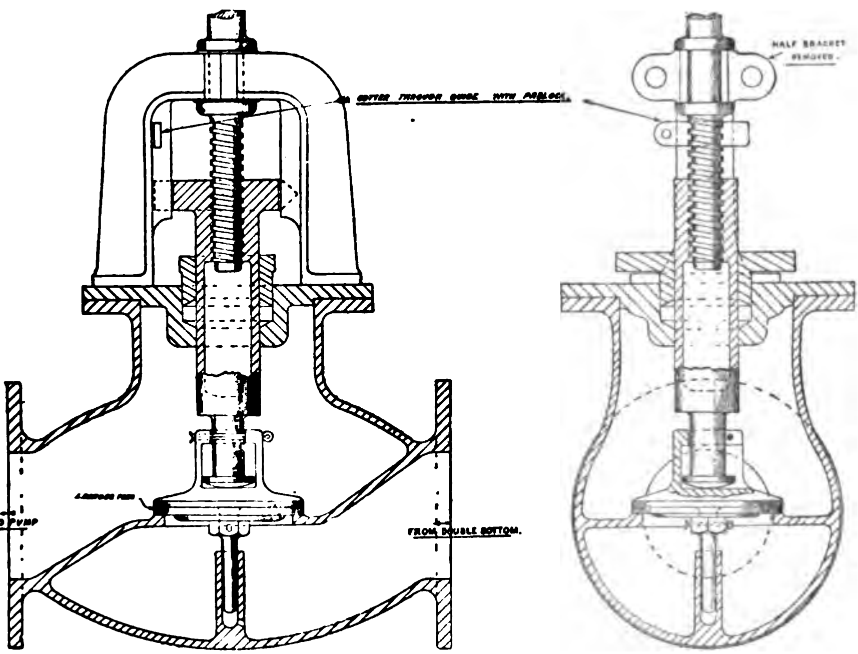


FIG. 342J.

be pumped overboard direct without contaminating the usual wash deck pump discharge pipes, &c.

(*e*) *Hand pumps.*—These have no direct communication with bilge as hitherto, and are only connected to the main suction pipe and sea.

In the ships referred to in Table II. the main drain and main suction pipes have both been omitted, and a steam or motor-driven pump is fitted in each main transverse compartment, in some cases the latter pumps being portable. These pumps are arranged so that they can deal with the double bottoms as well as the bilges, the suctions to the double bottoms being broken and fitted with hose connections. Watertight subdivision is still more relied on, and, as before men-

tioned, the main circulating pumps can only deal with leakage which enters the engine room direct.

Bilge suction pipes.—The following general principles are followed in fitting these pipes :—The bilge suction pipes of all pumps are of iron or steel zinced by the hot process, and are led to the lowest accessible part of the bilge, and provision is made for a free flow of water to each suction. In the case of the ordinary pipes in the machinery spaces the ends are carried to suction wells, F, in the inner bottoms, Figs. 342D and E, details of which are shown in Fig. 342F. These wells are of the *dished* form, not *wall-sided*, and the pipes are fitted from 2 to 4 inches below the level of the inner bottom and 6 inches above the bottom of the wells, which latter are about 2½ feet in diameter. Mud boxes M are fitted directly above the lower end of each of these pipes ; this length of suction pipe is straight, to allow the pipes being readily cleared if choked. Strainer boxes N, with portable covers, are fitted to the lower end of all suction pipes, and the plates are closely perforated with ⅜-inch holes, both at top and sides. The strainers are made about 10 inches high and about 2½ feet in diameter, in order to act as breakwaters when the vessel is rolling. The clear area through the holes in mud and strainer boxes is about three times the area of the suction pipe.

In long lengths of suction pipes non-return valves R (Fig. 342F) of light construction are fitted immediately above the mud boxes. These valves ensure the pipes being filled with water when not in use, in which case the pump has not to exhaust the air from the pipe before water can be pumped. Non-return valves are also fitted on all suction pipes through which sea water could be accidentally admitted to the ship, if this is not otherwise provided for on the system. No copper or brass pipes in the bilge are allowed to rest in contact with the hull work. All such pipes exposed to the action of the bilge water are well painted or varnished, and then covered with painted waterproof canvas.

Circulating pumps.—When in use for bilge purposes the circulating pumps generally discharge the bilge water through the condenser tubes in lieu of the cooling water obtained direct from the sea. In some cases separate pipes and in others valves are fitted in the condenser circulating water ends, so that in case the main engines are not at work the bilge water need not pass through the condenser tubes. If the pumps be reciprocating and worked by the main engines, which is still sometimes the case in the mercantile marine, their efficiency and power would be reduced in the same ratio as the revolutions of the engines, and under the circumstances the main engines could only be worked at reduced powers. If the circulating pumps be worked by separate engines, as is always the case in the Royal Navy, the above limitations would not exist, provided the circulating pumps could obtain sufficient cooling water from the bilge.

Circulating pumps, when worked by separate engines, are of the centrifugal type, and, for the small lift required to pump out the ship, the quantity of water that can be thrown by a comparatively small centrifugal pump is considerable. In the Royal Navy the efficiency of the circulating pumps for bilge purposes is always tested prior to the receipt of the machinery. In the latest battleships these centrifugal pumps are each from 3 feet 6 inches to 4 feet in diameter, and are

worked at a speed not exceeding 300 revolutions per minute. The numbers and capacity of these pumps are given in Tables I. and II.

Precautions to ensure circulating pumps drawing water.—Precautions have to be taken to ensure the efficient working of centrifugal pumps as bilge pumps. Ordinarily they circulate cooling water through the condenser; the inlet and generally the outlet orifice is below sea level, and the water runs by gravity to the pump. Centrifugal pumps cannot be relied on to draw water from any considerable depth, or in cases of single impeller pumps to deliver it to any great height, and to ensure that they will draw readily, in the excitement consequent on the ship making water rapidly, the pumps should be placed as low down in the ship as possible. It is also necessary that they should be fully charged with water at starting, and kept fully supplied when at work, for the presence of air in the suction pipes interferes with the action of the pumps, and any considerable quantity would stop them from pumping. Non-return valves (I, Fig. 342D) should therefore be placed at the bottoms of the suction pipes so that the whole length of suction pipes will be filled with water before starting the pumps to draw from the bilge. The suction pipe should also be of sufficient area to keep the pump fully supplied with water, without requiring too high a velocity in the pipe, usually about 9 to 10 feet per second. In some cases inefficient action of these pumps for bilge purposes has been due to the smallness of the suction pipe, the water not being able to enter the pump sufficiently fast to keep it fully charged. The pump suctions should be as low as possible and in all cases should be well below the levels of the furnace bars of the boilers.

Situation and power of engine.—Though the pump itself should be close to the bilge, the engine for working it ought, if possible, to be at a high level, so as to be out of reach of the water in case of its rising rapidly, as the full discharge of a centrifugal pump requires a high speed of revolution and no engine can work long with water surrounding it, turning the cylinders into condensers. This arrangement was carried out in several of the older Naval ships, the pump being placed in a horizontal position in the bilge, where it acts most efficiently, and worked by a vertical shaft carried to a considerable height and attached to the crank shaft of a horizontal engine. This, while very suitable for pumping efficiency, is generally inconvenient as regards the other engine-room fittings, and has not been repeated for some years.

Another point that should be kept in view is the provision of large engine power; for in the case of a serious leak circumstances may cause the pressure of steam to drop. For this reason the engines were made considerably larger than required for merely circulating water under ordinary circumstances, and were arranged to perform their specified output as bilge pumps with about two-thirds of the maximum boiler pressure. The valves for changing the suctions of the centrifugal pumps from the sea to the bilge are arranged to be worked in readily accessible positions, the wheels being kept at least three feet above the starting platforms. That too much care cannot be taken in arranging the above is demonstrated by instances on record in which ships seriously injured have been kept afloat long enough to run to harbour, or to enable the passengers and crew to be saved by the application of the circulating pumps.

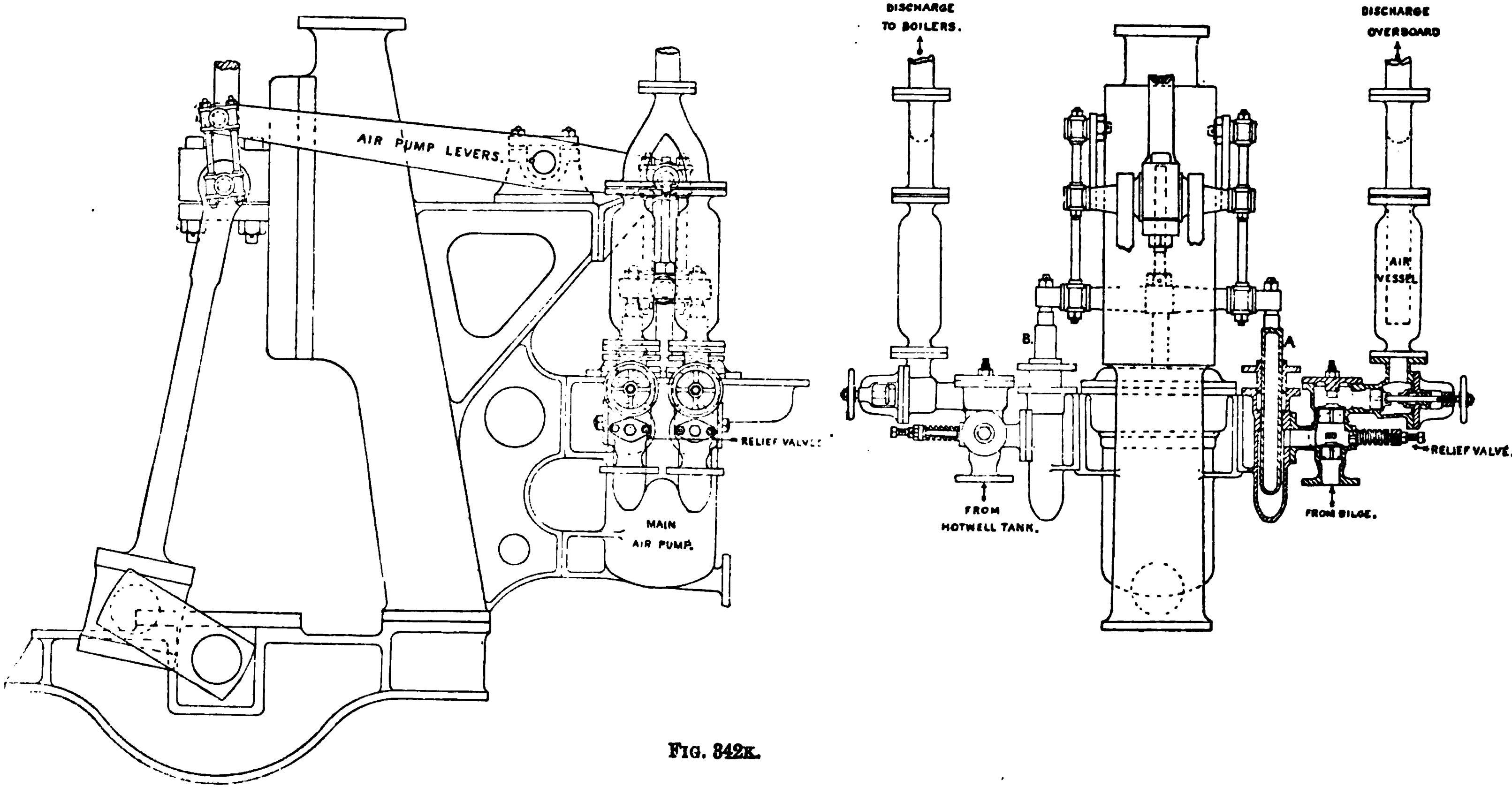

AIR PUMP LEVERS.
RELIEF VALVE
MAIN
AIR PUMP.
DISCHARGE
TO BOILERS.
DISCHARGE
OVERBOARD
AIR
VESSEL
B.
A
RELIEF VALVE.
FROM
HOTWELL TANK.
FROM BILGE.

FIG. 842K.

Main engine bilge pumps.—Formerly the majority of all steamships, including warships, were fitted with bilge pumps worked direct by the main engines, and this is still the common practice in the mercantile marine. Fig. 342K shows a general arrangement in which the pumps are worked from the main air pump levers, the bilge pumps A being fitted on one side and the boiler feed pumps B on the other.

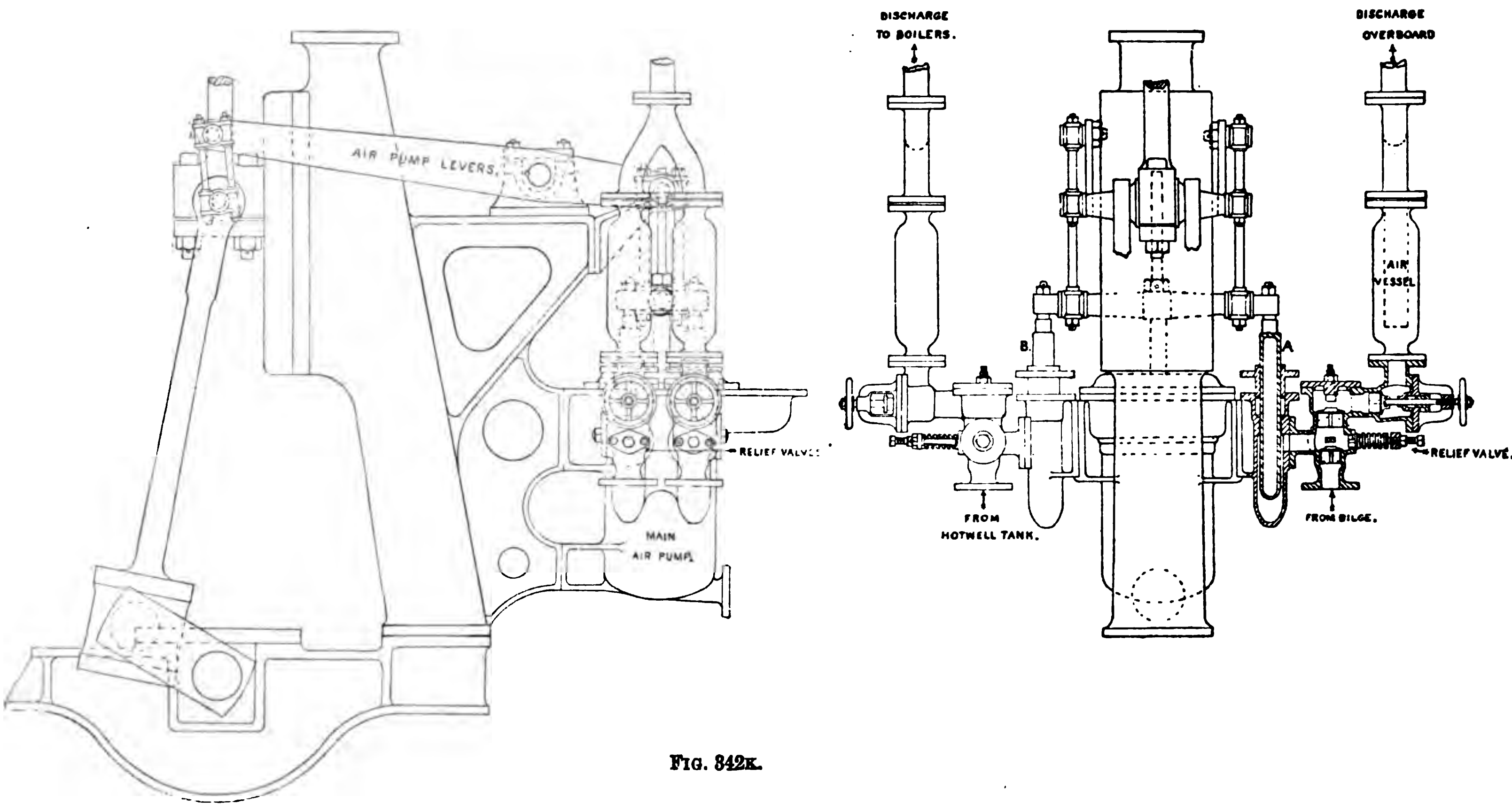
The bilge pumps are of the single-acting type and are not fitted so much to provide for extraordinary leaks as to clear the bilges of the water and oil that drains into them from pipes, bearings, &c., and other ordinary leakages. They are, however, usually much larger than is required for these purposes alone, so that where fitted their action should be taken into account in calculating the pumping power of a ship. The plungers of these pumps are always working with the main engines, and on this account they are not made as large as might otherwise be the case, as it would be difficult to keep the plungers and glands sufficiently lubricated to prevent overheating, and a considerable addition would be made to the constant friction of the engine, which is already large. Larger pumps would necessitate larger discharge pipes, valves, &c., which would be inconvenient and often impossible. In modern warships separate bilge pumping engines are fitted instead of pumps worked by the main engines.

A small save-all pump is still fitted in recent warships with reciprocating engines (Table I.), to assist in dealing with the water service, lubricating and other drainage. It is arranged to be worked either from the air pump levers or by an eccentric sheave or pin on the engine shaft, and also by hand when required. Its size is so small, however, as to be inappreciable as regards a leak in the vessel.

Separate fire and bilge engines.—The fire pumps and those for ordinary bilge pumping work are often pumps of different pattern. This has sometimes been the case in the Royal Navy, but for many years past in this service they have been of the same construction, and can be used for either of these purposes.

A sketch of one of these pumping engines, of the direct-acting type, with two cylinders and double-acting pumps, is shown in Fig. 342L. The mechanism is arranged so that the piston rod of each steam cylinder actuates the steam slide-valve rod of the other by means of the levers E and F. The motion transmitted therefore is positive and the slide valves follow the movements of the adjacent piston rods. These valves are generally fitted without lap or lead and the levers are arranged so that *when working*, both slide valves cannot be closed to the admission of steam at same time. Each engine is, however, mechanically independent of the other until steam is admitted. Levers G are provided for moving the pump rods and slide valves when engines are not in use.

The exhaust ports S S are separate from the steam ports P P, and the former are fitted nearer the centre of length of stroke in order to permit the piston to cut off the exhaust before the end of the stroke, and by cushioning the remaining exhaust steam, be brought to rest just previous to the admission of the steam, without striking the end of the cylinder. The area of exhaust ports is designed for a limited speed of the exhaust steam, and for this reason any attempt to force the speed of the pump will produce *short stroking* through excessive


AIR PUMP LEVERS.
MAIN
AIR PUMP.
RELIEF VALVE.
DISCHARGE
TO BOILERS.
DISCHARGE
OVERBOARD
AIR
VESSEL
B.
A
FROM
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FROM BILGE.
RELIEF VALVE.

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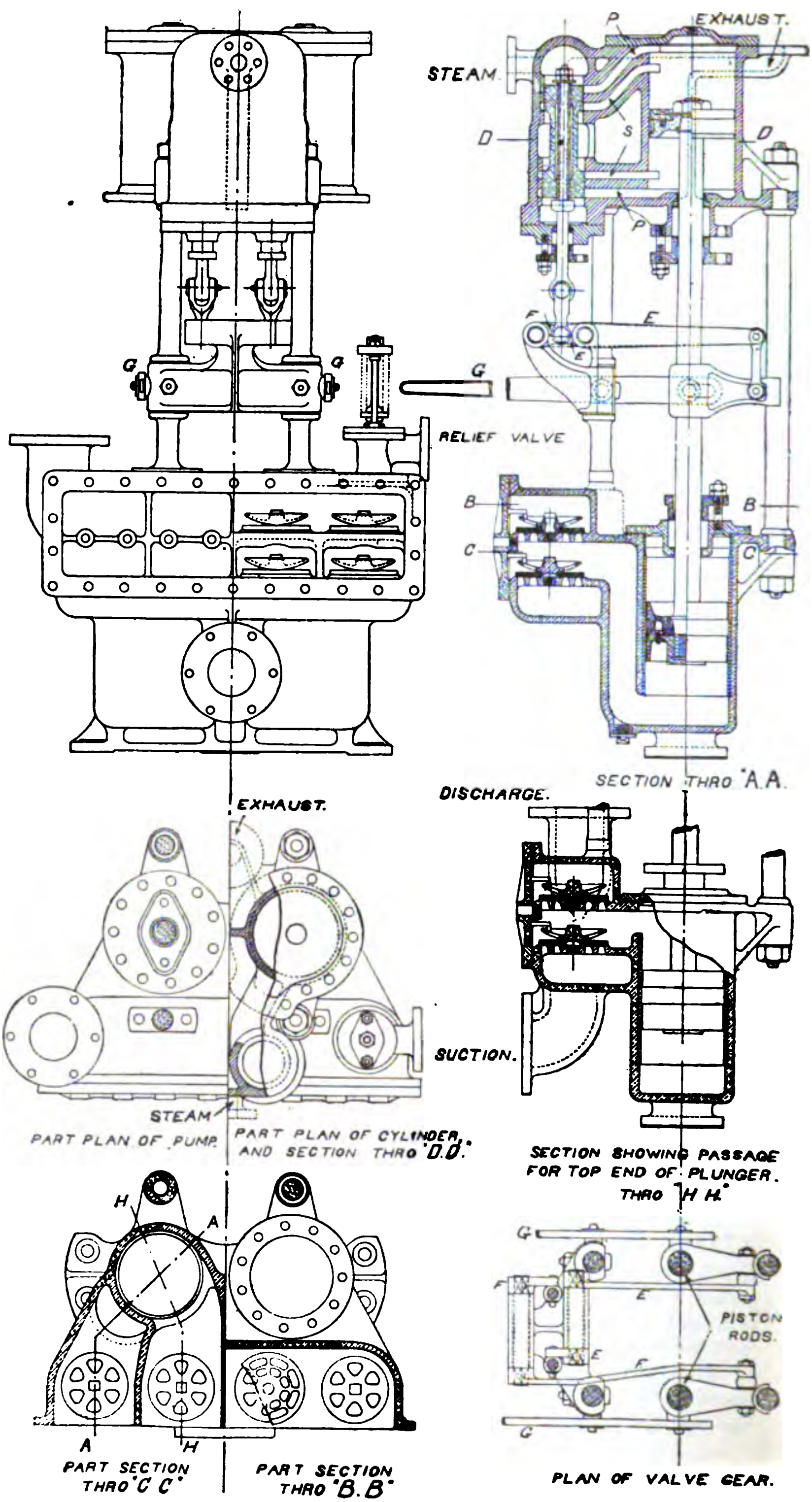

STEAM.
P
EXHAUST.
S
D
D
P
F
E
E
G
G
G
RELIEF VALVE
B
C
B
C
SECTION THRO "A.A."
DISCHARGE.
EXHAUST.
SUCTION.
STEAM
PART PLAN OF PUMP
PART PLAN OF CYLINDER, AND SECTION THRO "D.D."
SECTION SHOWING PASSAGE FOR TOP END OF PLUNGER. THRO "H H."
H
A
A
H
PART SECTION THRO "C C"
PART SECTION THRO "B.B"
G
F
E
PISTON RODS.
E
F
G
PLAN OF VALVE GEAR.

Fig. 342L.

cushioning due to fact that exhaust cannot get away sufficiently quickly. A similar result is obtained if the speed of the engine is suddenly increased through failure of the suction.

The good working of these pumps is much improved by allowing play between the collars on the slide valve spindle and valve. This device, though causing the operations of the slide valve to be somewhat later in action at the initial stages, keeps the steam and exhaust ports wide open longer than would be the case if the rod were rigidly attached to the slide valve. The former gives a more uniform steam pressure on the piston and the latter allows a freer exhaust, and this tends to prevent short stroking. The sketches indicate the construction and action of the pump end. Besides the type of pump above described crankshaft pumps are also in common use for this purpose.

At least two such pumps are fitted in all except the smallest warships; the numbers and capacities being as in Tables I. and II. preceding. The pumps are large enough to remove the stated quantities with revolutions or double strokes not exceeding 60 and 35 per minute for crankshaft and non-crankshaft pumps respectively, and with steam pressure of about two-thirds the maximum boiler pressure. Double cylinders are desirable for these engines to facilitate starting; and the slide valves are made with little or no lap, to ensure the engines starting readily in any position, economy in the use of the steam being in these cases a minor consideration. The pump barrels are fitted with separate liners to facilitate renewal; a relief valve, air vessel, and a pressure gauge, are always fitted in connection with the pump delivery. Air valves are fitted in the highest parts of all water suction passages in the pump casings in which air is likely to be imprisoned.

Latrine pumps.—Separate steam pumps are sometimes fitted for latrine purposes in a few modern large warships. These 'latrine pumps' discharge sea water into the fire main direct for use in connection with the sanitary system.

Steam ejectors.—The vessels of the torpedo flotilla have no communication through the transverse bulkheads, and to reduce the number of pipes passing through each bulkhead to a minimum, each main compartment is fitted with an ejector. In the larger destroyers each ejector has a capacity of 40 tons per hour. A regulating discharge valve is fitted which can be shut down and steam can be directed back through the suction pipe to clear the latter in case it should be choked. Ejectors are wasteful in steam, but are much lighter and occupy less space than a steam pump, which is of special consideration in these vessels. Small steam ejectors, with a single orifice, are fitted to steam launches, pinnaces, &c. The steam pipe is $\frac{1}{2}$ inch to $\frac{5}{8}$ inch diameter, with $\frac{1}{4}$ inch orifice, and the discharge pipe is $\frac{3}{4}$ inch to $\frac{7}{8}$ inch diameter. Such ejectors have been found by experiment to be sufficiently powerful to force water out of the bilges faster than it would flow in when a plug, about 1 inch diameter, was removed from the bottom of the boat.

Hand pumps.—These pumps of the Downton type are generally fitted below the water-line, but are actuated by gearing and temporarily fitted cranks carried in bearings secured in columns on the decks above. These cranks are situated high up in the vessel, and require from 4 to

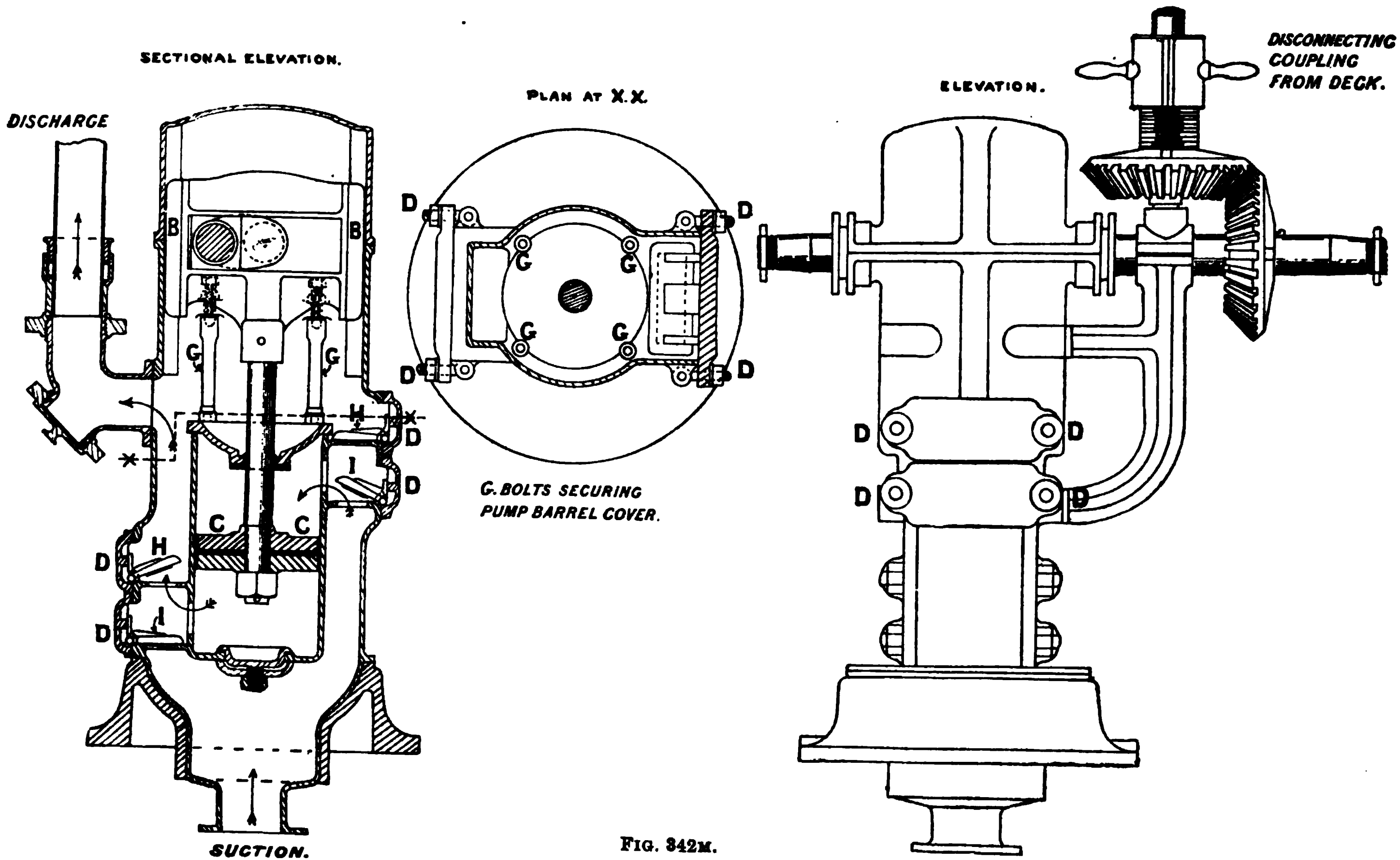

SECTIONAL ELEVATION.
DISCHARGE
SUCTION.
B
B
G
G
C
C
H
H
I
I
D
D
D
D
PLAN AT X.X.
D
D
D
D
G
G
G
G
G. BOLTS SECURING
PUMP BARREL COVER.
ELEVATION.
DISCONNECTING
COUPLING
FROM DECK.
D
D
D
D

FIG. 342M.

30 men to operate them, according to the size of the pump. They are made in the following four standard sizes: diameter, 5 inches, stroke, 5¾ inches, diameter either 6 or 7 inches, and stroke 6 inches; diameter 9 inches, and stroke 7 inches. Fig. 342M shows the details of Messrs. Stone's pump; the motion of the crank-pin is converted into reciprocating motion by the crosshead B B, which is guided vertically by grooves in the body of pump and connected to the leather-packed plunger C. The pump is double-acting, and the valves, of which there are four, are of metal, faced with leather. The suction valves I and discharge valves H are readily accessible for examination, doors D D being fitted for this purpose. The working parts are completely closed in and require little attention.

Special pumping fittings.—Large ejectors of the Friedman type were fitted in several of the earlier armourclads to deal with water which might enter the bunker spaces. The pulsometer, an arrangement by which the direct pressure of the steam acts on the water, has also been fitted in many mercantile steamships, generally for use in connection with ballast tanks. With both these designs large quantities of steam are lost, and their use has been discontinued on board ship.

In addition to the pumping arrangements previously described pumps are provided for the following purposes: Pumping fresh water for ordinary purposes from the storage tanks to the galleys, filter tanks, wash places, etc.; pumping sea-water for sanitary purposes, wash places, etc. These pumps were formerly of the Downton or ordinary lift type, and were worked by hand, but are now usually driven by electric motors.

Flooding arrangements.—Arrangements are fitted in warships so that the spirit room and all magazines can be flooded direct from the sea in case of fire. Generally a group of these compartments is flooded by means of one master valve, which is directly connected to a sea-valve used for some other purpose. In addition a shut-off regulating valve is fitted to each compartment. The pipes are generally from 3 inches to 6 inches in diameter, according to the size of the compartment, and in some cases of very large magazines two flooding pipes are arranged for. Each flooding valve spindle is fitted with a special locking arrangement, A, Fig. 342N. This sketch indicates the method of actuating the index which shows the open or closed position of the valve on the deck plate, by means of a nut B which is guided in vertical grooves C and has projections E which work in the spiral grooves D of the indicator tube. Similar methods of actuating indexes are fitted for the water-tight door spindles, but in these cases the locks are omitted.

Flooding valves are always fitted to close with a right-hand motion. Automatic arrangements are made for letting the air out of the compartment when flooding. When the roof of the compartment is well below the water-line an automatic valve, Fig. 342O, is fitted. A wire gauze diaphragm is fitted as shown, the portion of the pipe immediately above it being readily portable to facilitate the examination and replacement of the wire gauze fitted. The outlet pipe extends to the deck above the magazine crown, and has its end turned over and closed, and the upper part of the bend is pierced with small holes of ample combined

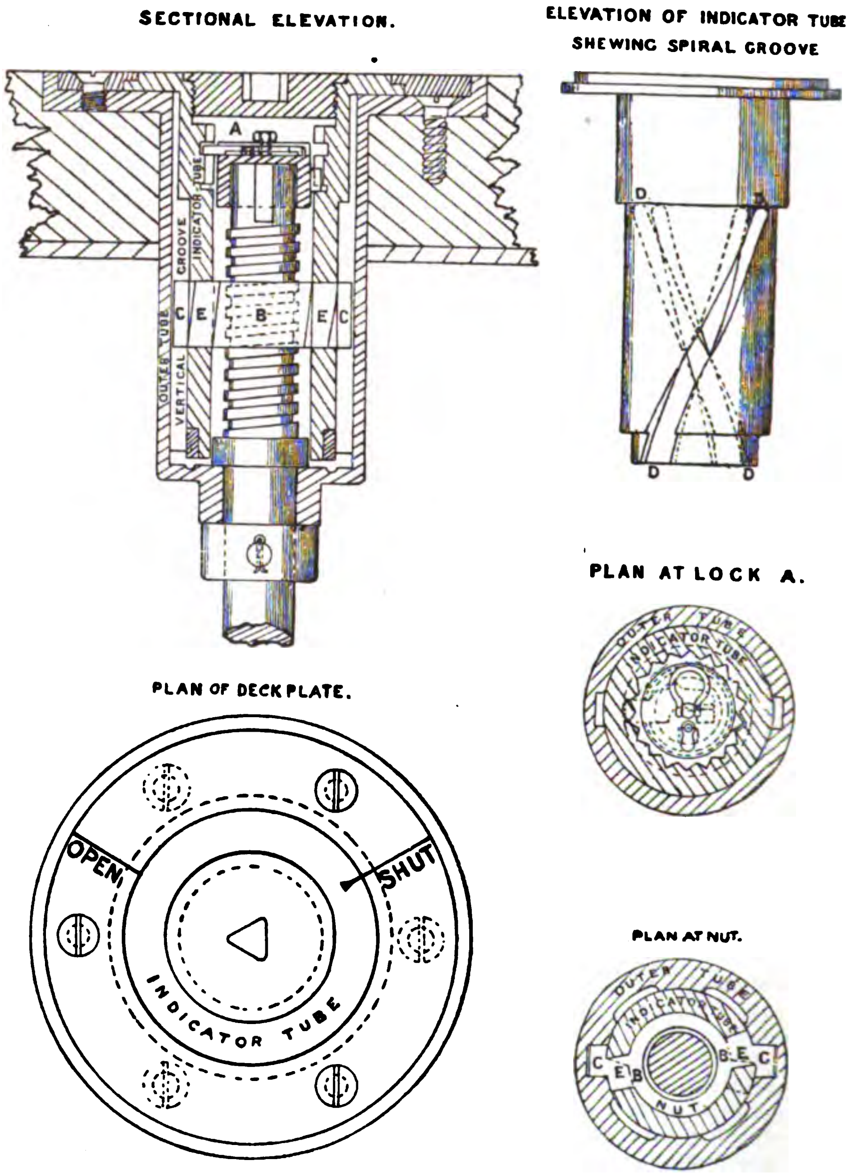

SECTIONAL ELEVATION.
A
INDICATOR TUBE
GROOVE
C E B E C
OUTER TUBE
VERTICAL
ELEVATION OF INDICATOR TUBE
SHEWING SPIRAL GROOVE
D
D D
PLAN AT LOCK A.
OUTER TUBE
INDICATOR TUBE
PLAN OF DECK PLATE.
OPEN
SHUT
INDICATOR TUBE
PLAN AT NUT.
OUTER TUBE
INDICATOR TUBE
C E B B E C
NUT

FIG. 342N.

area. Where the crown of the magazine is near the water-line, the valve is omitted, and the exhaust pipe is led above the tops of the watertight bulkheads. The upper and lower ends are fitted with wire gauze, the lower being protected with a strong brass or copper rose, and the upper with a sliding watertight gunmetal shutter. Shell rooms are not usually fitted with special flooding arrangements except in the most recent ships, but stop valves with hose connections are frequently provided on the flooding pipes so that they can be flooded

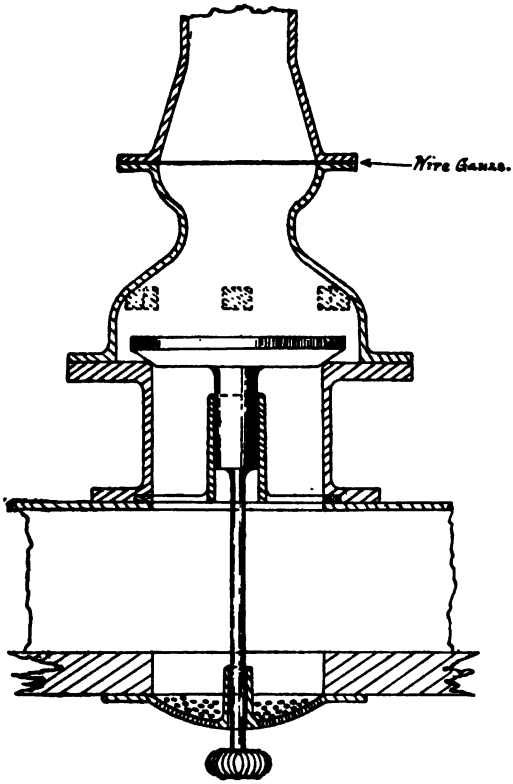


FIG. 342o.

by hoses if necessary. They are generally emptied as in the case of magazines.

For restoring trim and correcting heel it is necessary that water may be quickly and readily admitted to the double bottom and wing compartments. The double bottom compartments are flooded either direct by hoses or through a branch connection from one of the sea-valves for the pumps, a master locked valve being fitted on the sea valve for this purpose.

In ships where the pumping etc. arrangements as described on p. 371g are fitted, the double bottom compartments are flooded through the special valve on their branch connections to the main suction pipe

(Fig. 342J), the main suction pipe being in connection with the hand-pump sea-valves through a locked valve. In the ships detailed in Table II. all the double-bottom compartments under the main machinery spaces are appropriated either for the stowage of oil fuel or for reserve feed-water, and no special provision is made for flooding, but if necessary they may be flooded through portable hoses rigged on fire-pump discharges. The other double-bottom compartments, i.e. those forward and aft of main machinery spaces, are flooded through the sea-valves of the electric motor-driven pumps. Directing boxes with screw-down valves and suctions to each double-bottom compartment are fitted in each main transverse compartment, and each directing box is connected to the pump sea valve through a flooding valve of the type referred to above.

In most of the modern ships provision is made for flooding the wing compartments; for those abreast machinery spaces special sea-valves are fitted, sluice valves being arranged as necessary between the various adjacent compartments. The wing compartments abreast the double bottoms beyond the machinery spaces can in some instances be flooded direct through the main drain pipe, since non-return valves are not fitted on the main drain branches to these compartments. The wing compartments of ships referred to in Table II. can be flooded through branches taken direct from the sea valves of the motor and steam bilge pumps, sluice valves being fitted between compartments.

Dry dock flooding arrangements.—It is necessary to provide for the contingency of fire when the ship is in dry dock. In the older ships for this purpose a group of hose connections with a shut-off valve at each group are fitted on the upper and between decks, the latter to provide for the possibility of a fire making the upper deck untenable when it might be considered necessary to flood the magazines. These hose connections enable water to be led from the shore to the flooding pipes of the ship. In the latest ships, to effect this, provision is made so that when the vessel is in dock a flange with a hose connection can be attached to the outboard sides of the inlet branches of the flooding sea-valves, the gratings over these orifices being removed for this purpose. Hoses can then be connected to the dockside hydrants.

Fire main.—All the steam fire and bilge pumps, except any worked off the main engines, the steam latrine pumps, the bilge electric motor, and the hand pumps are fitted with sea suctions, and are arranged to deliver at the necessary pressure into a pipe called the fire-main, which is carried fore and aft in the ship, with rising mains and branches leading to different parts as required. Delivery valves with hose connections are fitted to these rising mains to which hoses may be connected. These connections are also used for wash-deck purposes, &c.; non-return valves are fitted at the junctions of delivery pipes from pumping engines with the fire-main. The main run of fire main pipe is carried below the protective deck, and arrangements are made by means of special shut-off valves fitted at the base of each rising main and workable from the decks above as well as at the valve itself, so that in case of an accident to the rising main, that special section can be shut off and the defect be localised. To ensure a supply of water for the compartments above the protective deck in the event of the rising mains being thus disabled, additional valves with branches are taken from the

main below the protective deck. The branches are carried through the protective deck, and there terminate in a screwed socket, to which a goose neck with hose connections can be attached from above. The valve can be worked either from above or below, and the orifice is ordinarily closed by a deck plate, flush with deck.

For dealing with small fires, small bib-valves are fitted to the rising mains to enable water to be quickly drawn off in buckets, thus avoiding delay in dealing with the fire while a hose is being rigged. Branches are also provided on the fire-main for providing water for the sanitary arrangements; for washing out boilers and guns; for running through hawse-pipes when working cables; and for capstan and other auxiliary engine water service. Connections are generally provided so that the steam pumps delivering to the fire-main can be utilised for circulating water through the condensers of refrigerating and ice-making machinery, and the cooling jackets of oil-driven dynamos in case the separate circulating-water pumps of these machines become defective; also for drenching the ammonia parts of ice-making machinery where such are fitted.

In the ships referred to in Table II. the special fore and aft main run of fire-main under protection has been discontinued, and the pumps in each compartment deliver direct to the various decks immediately above the compartments in which they are situated. The rising mains from the separate pumps are, in some instances, connected above the main deck, and this, in a measure, takes the place of the usual fore and aft run.

CHAPTER XXVIII.

AUXILIARY MACHINERY AND FITTINGS.

In modern steamships, especially in those of the Royal Navy, the auxiliary machinery and fittings are of great importance. In H.M.S. 'Powerful,' for example, there are ninety-nine different auxiliary steam-engines for various purposes in addition to the main engines of the ship. In the battleships of the 'Majestic' class there are 72 auxiliary steam-engines and 32 hydraulic turning engines, lifts, bollards, &c. Those vessels are really huge floating war machines, where all the principal operations for working, steering, and fighting are performed by steam or hydraulic power with little manual labour, so that the vessel's efficiency will depend largely on the condition of the machinery department.

The enumeration of the various kinds of work on board ships that are now done by steam power would be sufficient to show the importance of this part of the duty of the engineer. In addition to the main propelling engines, steam power is used for ventilating the ship and supplying air to the boilers; weighing the anchor; steering; pumping; working turrets, and loading, training, and working the guns; compressing air for charging and launching torpedoes; putting torpedo and other boats into and lifting them out of the water; distilling fresh water; producing electricity for lighting the vessel, for search lights and driving motors; for refrigerating purposes; for actuating the workshop machines; lifting coal into the vessel, &c.

Steering engine.—Figs. 343 and 344 show an arrangement of steam steering engine, which is found by experience to answer all requirements. An ordinary two-cylinder engine drives a main shaft A which leads to the tiller by suitable worm or toothed gearing. This gearing reduces the speed of the shafting near the tiller, and so increases the turning movement available to overcome the resistance of the rudder when the ship is proceeding at high speeds.

The engine contains an arrangement by means of which it only moves when the steersman moves the steering wheel on deck. The rudder then remains fixed at the required inclination till the steersman again moves the steering wheel. Details of the gear by means of which this is accomplished are given in fig. 345. The valve B is a reversing valve similar to that shown in fig. 181, and it is worked by means of shafting C led from the steering wheels on deck at various parts of the ship. The motion of this shaft, by means of the pair of spur wheels D, rotates the screwed shaft E, which therefore moves up or down in the nut F, a groove and feather fitted at the small spur wheel enabling the shaft to slide. The nut F is prevented from moving in the vertical direction, so that the motion of shaft E, attached to the

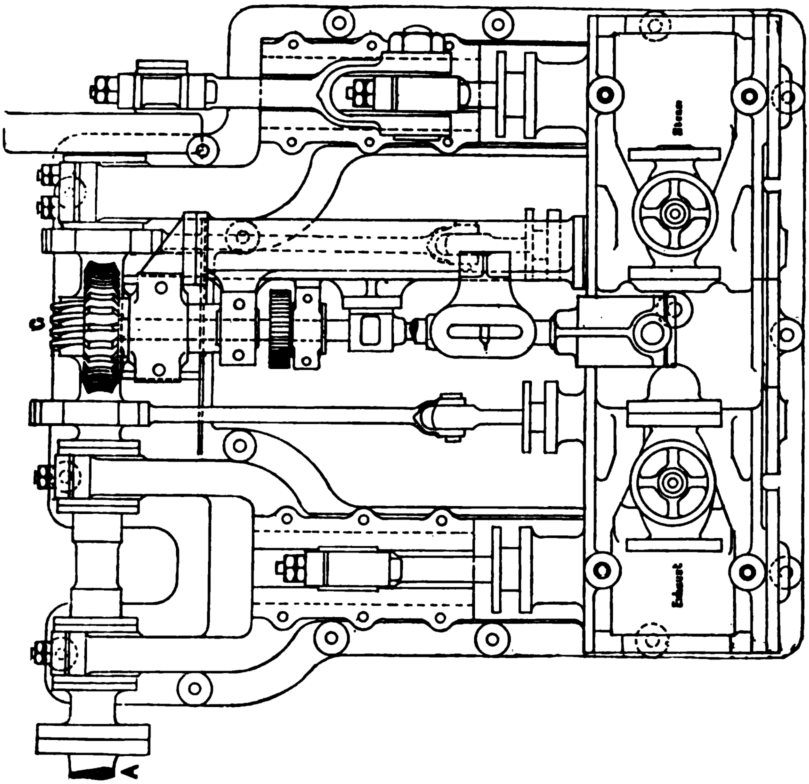


FIG. 344.

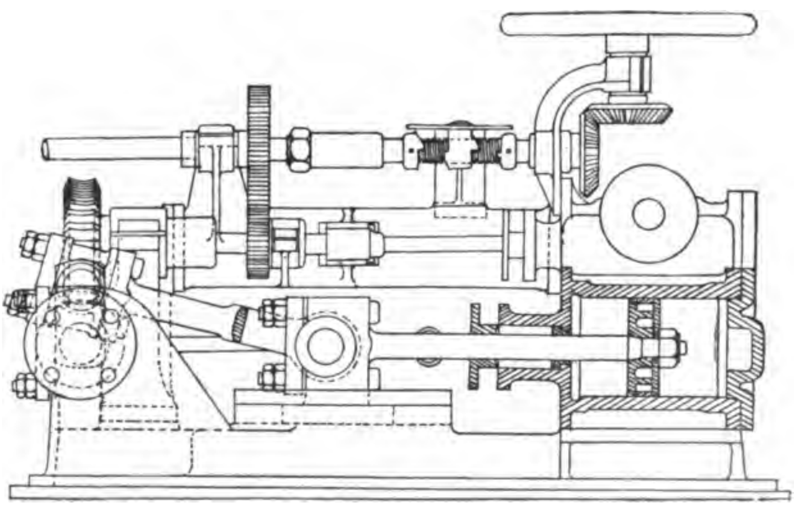


FIG. 343.

reversing valve rod by means of a revolving nut H, opens and shuts the reversing valve.

On the engine crank shaft there is a worm G (fig. 343) which engages with a worm wheel formed on the nut F, so that this nut revolves with the engine and causes upward or downward motion of the reversing valve. The gear is so arranged that when the valve is opened by the motion of the steering wheels on deck, the consequent motion of the nut F brings the valve back again to the closed position. When therefore the steersman opens the valve and starts the engine in the direction necessary to put the rudder either to starboard or port, the engine replaces the valve in the central position and so stops. Consequently the engine only works while the steering wheel on deck is being worked, and when this wheel is not moved further, the engine and therefore the rudder remain fixed in whatever position it has been moved to, until the steersman again moves the steering wheel.

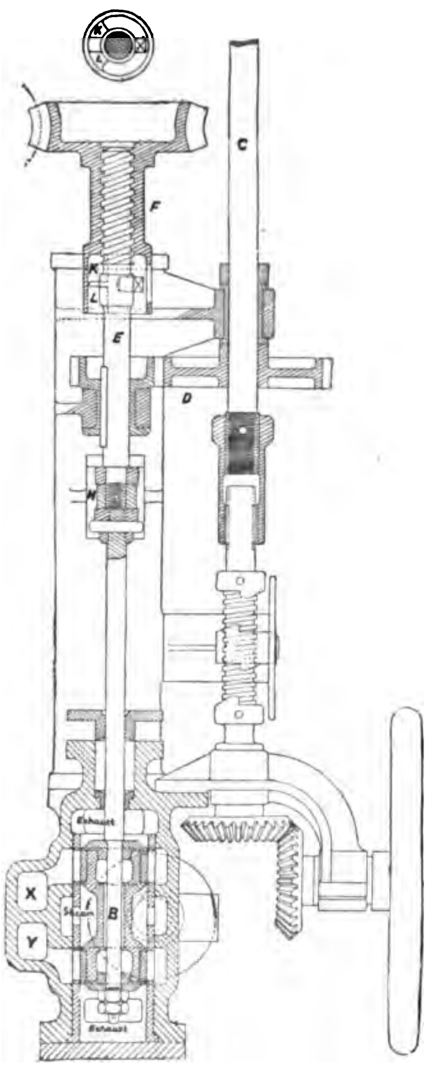


FIG. 345.

Two stops, K, L, are formed in the nut F, which engage with a pin on the shaft when the latter has made half a revolution from the middle position in either direction. This prevents the steersman opening the reversing valve too wide.

Steam enters the reversing valve casing at the centre, while the ends are connected together by an outer passage leading to the exhaust pipe. When the reversing valve is moved upwards the passage X is supplied with steam while Y exhausts steam, and *vice versâ* when the reversing valve is lowered. The passage X leads to the centre of the *engine slide valves*, while Y leads to the middle of these slide valves, which are of the piston type. This interchange of steam and exhaust reverses the engine.

Instead of transmitting the motion of the steering wheel to the

engine by means of shafting and bevel wheels, *Brown's 'Telemotor'* is fitted in several recent warships of the Royal Navy, in which the motion is transmitted by water pressure in small pipes, a certain motion of the piston of a water cylinder, called the 'transmitter,' at the steering position on deck, producing a similar motion in the piston of a water cylinder, called the 'receiver' cylinder, near the steering engine, by means of which the reversing valve is moved by a simple form of lever, so arranged that the resulting motion of the engine works the engine controlling valve back to its middle or 'stop' position. Means are provided for preventing inaccuracy due to changes of volume of the fluid through differences of temperature or losses from joints, and for reducing inaccuracy from leakage past the pistons.

Steam steering wheels. — The wheels which work the reversing valve of the engine, called steam steering wheels, one of which is shown in Fig. 346, are fitted on pedestals which contain stops to prevent the wheels being turned too far in either direction, which would cause the rudder to be placed at too great an angle, and strain and endanger the steering gear. The number of turns of the steering wheel from hard a port to hard a starboard is arranged to be eight, and more revolutions than this are prevented by projections on the sliding nut A, which, when the maximum number of turns have been made, engage with corresponding stops pinned on the shaft, so that the wheel cannot be moved further. The wheel also works an index finger on the top of pedestal, which shows the steersman the position of the tiller.

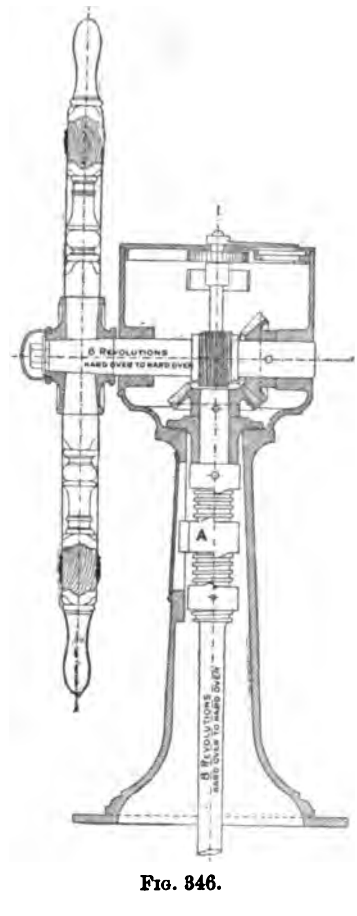


FIG. 346.

Gear between engine and rudder. —The steering engine sometimes works, through its shaft, a barrel on which wire ropes are wound, which lead to the end of the tiller as in the ordinary hand steering gear. This, however, is not so suitable for steam power, as there is risk of the ropes being strained if the engine be moved quickly; and the holes in the bulkheads of the ship through which the ropes pass cannot be made watertight.

In all gear worked by steam power, the arrangements should be as rigid and mechanical as possible, and shafting and gearing be adopted if practicable, in preference to ropes or chains. The holes in the bulkheads of the ship through which the shafting is carried can then

be made watertight by suitable stuffing-boxes. More generally, therefore, and in all warships except those of the smallest class, shafting is carried from the steering engine to the after part of the ship, where it actuates the rudder by means of suitable gearing.

Details of rudder gears.—One arrangement which has been found efficient is 'Rapson's slide,' largely used in battleships, and shown in Figs. 347 and 348. With this arrangement there is a sliding frame, travelling on guides or rails running transversely across the stern of the ship, and the slide is pulled on one side or other of the middle line by a pitch chain, A, having tightening screws, and actuated by the steering engine through a 'sprocket' or chain-working wheel, B, below the slide. The sprocket is rotated by the steering engine or hand steering gear, whichever is connected.

The end of the tiller, C, is parallel and of rectangular section and a swivelling block carried by the sliding frame fits over it. As the steering engine moves the sprocket wheel, the sliding frame moves along the guides and carries with it the end of the tiller, which slides in the swivelling block, the latter rotating slightly in the horizontal plane to suit the angle of the tiller.

The arrangement shown is that of a modern battleship with two steering engines and hand steering gear; the wheels and clutches shown explain how either steering engine or the hand gear can be used to operate the rudder. By turning the small wheel shown in connection with the clutch levers the three slots are turned round, and these are so arranged that only one clutch is in gear at the same time. The wheels are loose on the shaft, while the clutches work on feathers, and the one in gear drives the steering shaft.

Another arrangement, which is designed to increase the turning moment on the rudder as its resistance increases, is Harfield's compensating steering gear, shown in Figs. 349 and 350. In this gear the engine drives a horizontal bevel wheel, A, loose on the vertical spindle, but which, by means of a clutch keyed to the vertical spindle, rotates an eccentric pinion, B, on a vertical shaft. This eccentric pinion gears with a rack, C, so shaped as to always engage with the eccentric pinion, while moving about a fixed centre. The force is transferred to the rudder by means of double rods which simply transmit the force from the fixed axis to the rudder-head, but do not alter it, so that when examining the action of the gear the fixed centre may be considered to be the rudder-head.

Assuming the engine to exert a constant turning moment on the vertical shaft, it will be seen that when the rudder is central the longer radius of the eccentric pinion acts on the rack, and the motion is comparatively rapid and the force exerted small. As the rudder increases its angle, and therefore its resistance to motion, the smaller radius of the eccentric pinion comes into operation, and the force exerted is correspondingly increased. A worm leading from the hand steering position, gearing with a worm-wheel D, also loose on the vertical spindle, are provided to enable hand power to be also used, the clutch being the means of transference from steam to hand power. To prevent risk of accident when this transference takes place it is necessary that the rudder pins should be shipped, so as to secure the rudder. These are shown in Figs. 349 and 350.

Hydraulic steering gear.—In many ships hydraulic power has

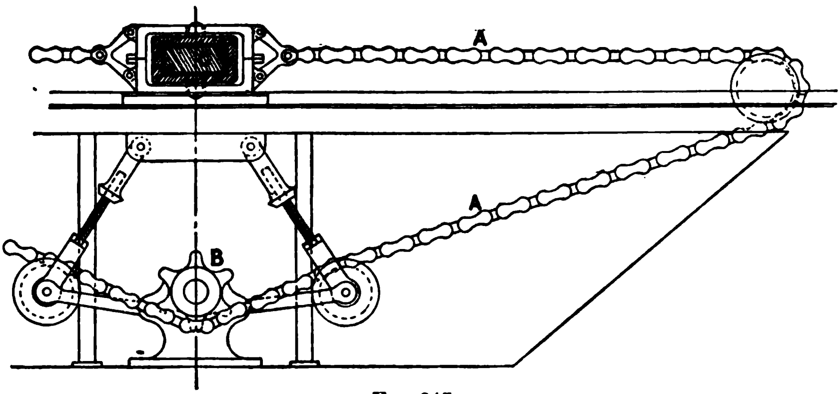


FIG. 347.

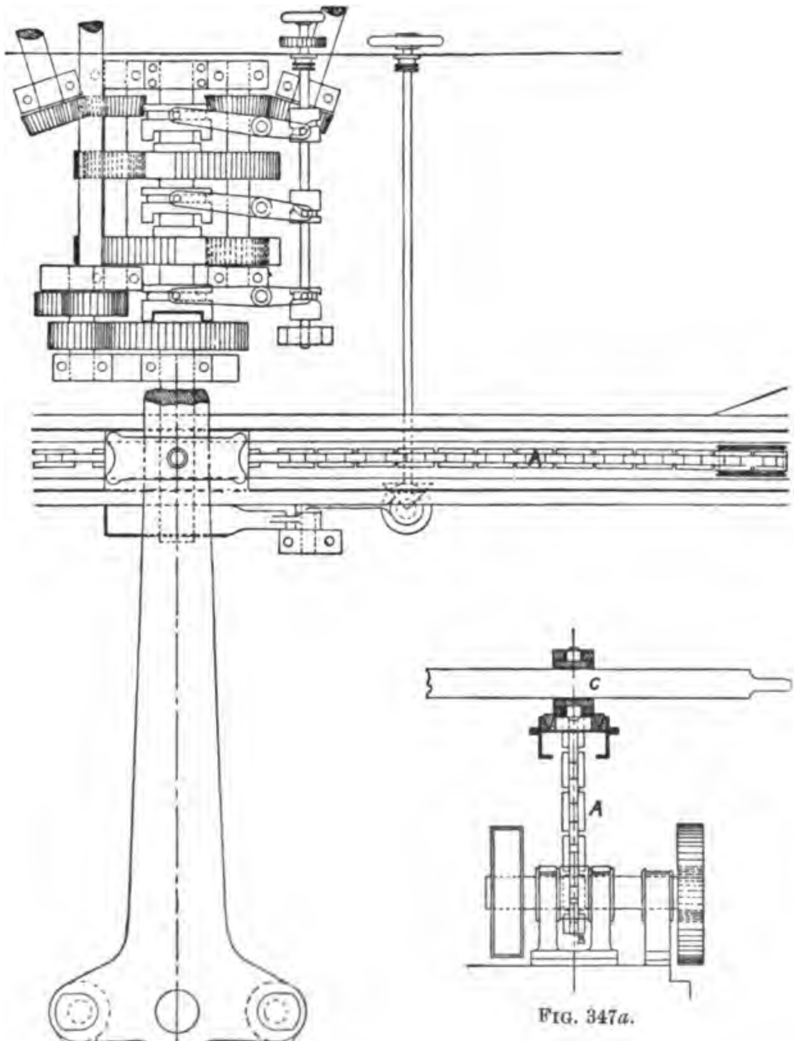


FIG. 347a.

FIG. 348.

been used, instead of steam power, for working the rudder, and various plans have been adopted for utilising water pressure for this purpose. In several ships that have been fitted with this kind of steering gear, the hydraulic pressure has been utilised for other auxiliary purposes, such as for working cranes, winches, reversing gear for main engines, &c.

The advantages of steam or hydraulic power for steering in the case of large ships need no discussion. It places the control of the ship practically in the hands of one man, who is enabled to place the rudder hard over to starboard or port, when the ship is steaming at full speed, with little exertion, perfect safety, and in very much less time than would be required with hand gear, even when worked with a large number of men. It is, however, essential that the steering apparatus should be simple in construction and arrangement, and safe and reliable in operation, so that much skill, care, and attention are required to be devoted to the design of the whole of the details of the gear to reduce complexity and promote efficiency to as great an extent as possible.

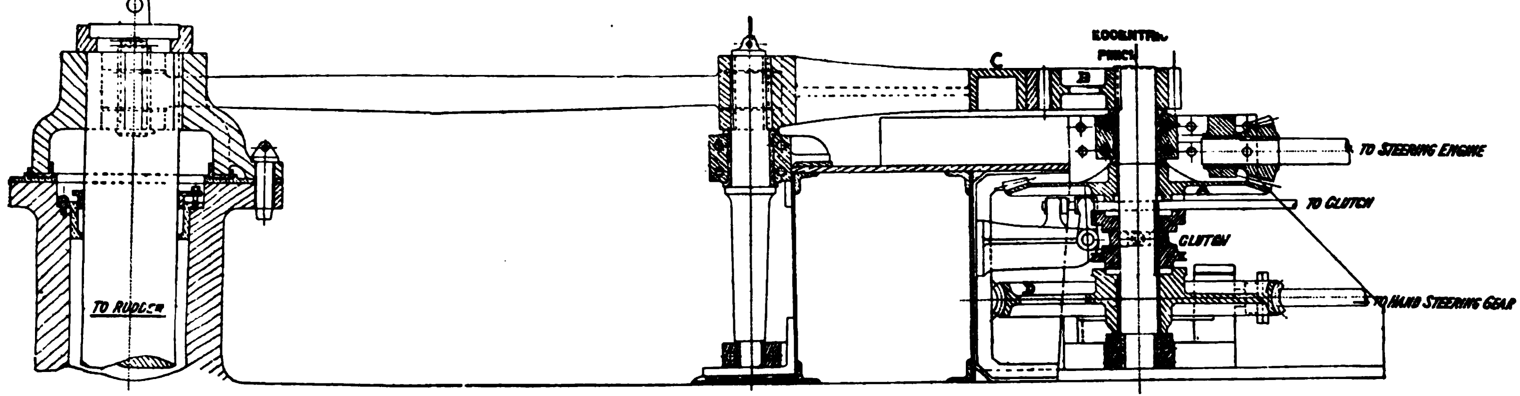
Capstan engine and gear.—The capstans of steamships of any considerable size are worked by steam, the power being usually applied through the medium of a worm-wheel, keyed on the lower end of the spindle, which is driven by a worm worked generally by an engine fitted for this purpose only. A large warship would be fitted at the forward end with duplex cable holders for working $2\frac{3}{8}$-inch cable, one on each side of the ship, with a combined warping capstan and cable holder at the middle line. Each of these spindles is carried through the decks to the capstan engine flat, where the worm-wheels are fitted on their spindles, a suitable bushed step being provided in the bed-plate for taking the weight of the spindle and cable holder or capstan.

The worms driving these worm-wheels are generally worked from the crank shaft of the capstan engine by steel mitre gearing with helical teeth, arranged with suitable clutches so that the cables may be both hauled in or both veered, or one hauled in and the other veered, simultaneously, without reversing the engine. In the case of the centre forward capstan, in order to meet the case of working by hand by means of capstan bars, the worm-wheel is loose on the spindle and driven by a disc keyed to the shaft and placed immediately above the worm-wheel. The clutch-gear provided at the capstan engine enables the direction of the cable holders to be reversed.

The capstan engines are fitted with reversing valves, which may be actuated from the cable working deck, so that any desired motion can be obtained. When the capstan is being worked by steam power the safety pawls fitted for hand working should be kept securely lifted to prevent accident, the gearing of the engine itself being sufficient to hold the capstan in any position.

Cable holders.—These are generally of cast-steel and arranged to run loose on their shafts, as shown in Fig. 351. They are fitted with friction plates of wrought-steel and brass, shown in the lower part of the figure, and cast-steel driving-discs keyed firmly to the spindles. The friction plates are actuated by a cast-steel compressing nut, so that by turning this nut in the proper direction it gives compression to the friction plates, and so enables the driving disc to work the cable holder.

Fig. 349.



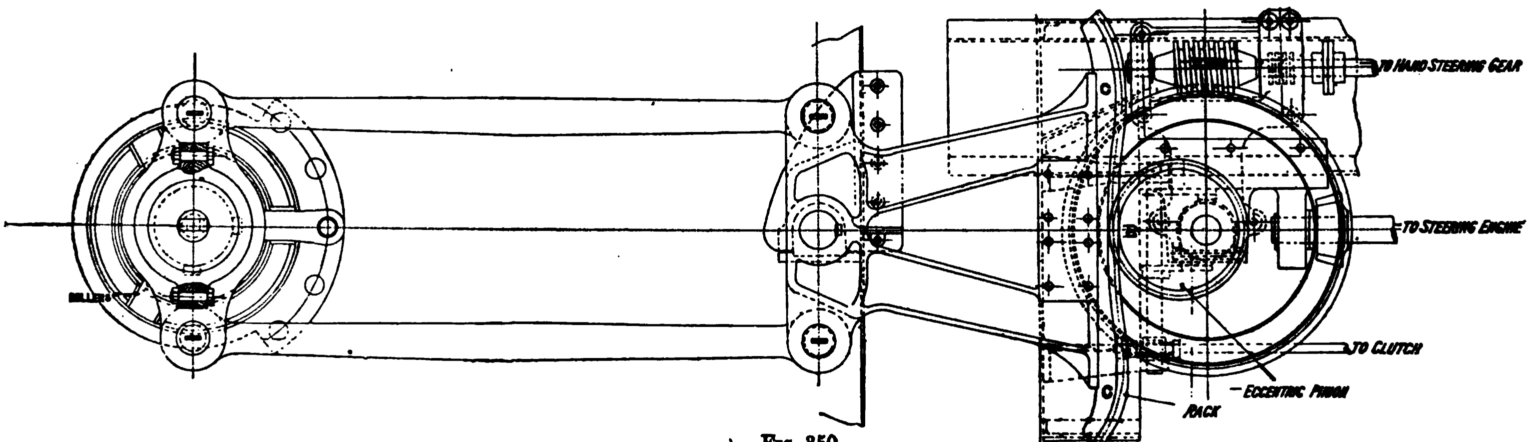


Fig. 350.

The centre line capstan cable holder, Fig. 352, is securely keyed to the spindle, and is provided also with a smaller cable holder above it, for $1\frac{1}{8}$-inch chain to perform the operation of catting the anchor. Means are also fitted to enable this centre capstan to be worked by hand, either from the upper deck or some intermediate one.

An *after capstan* and engine is also fitted in the largest vessels in a similar manner to the forward capstan, and suitable for working a smaller size of cable, say $1\frac{5}{8}$-inches. Portable whelps for working steel-wire hawsers are also fitted, one of these being indicated in position at the left of Fig. 352.

Limitation of force exerted by capstan engine.—The cylinders of all auxiliary engines should be made large enough to enable the engines to be efficiently worked with reduced pressures of steam. In the case of the capstan engine, it is therefore necessary to prevent the possibility

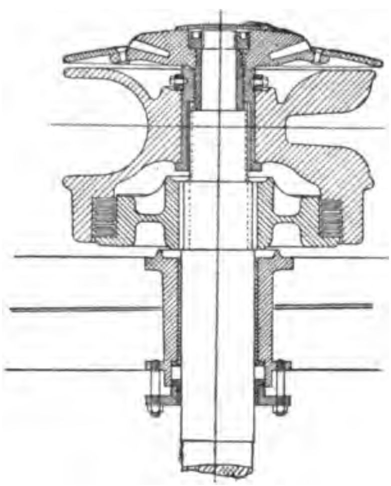


FIG. 351.

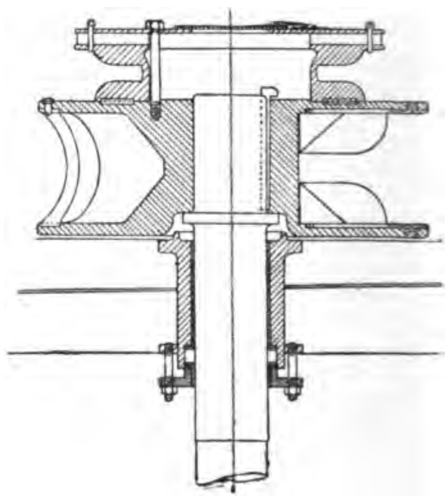


FIG. 352.

of excessive strains being brought on the chain cables when higher steam pressure is being used. The maximum pressure to be allowed in the steam chest of the engine is such that a pull of more than two-thirds of the proof-strain of the cable cannot be exerted by it. To effect this, reducing valves are sometimes fitted, but more generally in the Royal Navy a pressure gauge with prominent mark showing the maximum pressure, is fitted on the engine side of the stop-valve, so that the attendant can regulate the opening of the windlass stop-valve accordingly.

The use and advantages of hydraulic machinery.—Hydraulic power is largely used in battleships for working the heavy guns. For this purpose it has many advantages. The principal reasons for its adoption are :—

1. It is noiseless and steady in working.
2. It is easily controlled over a large range of speed.

3. It is always ready for use, no preliminary preparation such as warming, being required.

4. As the parts of the machinery are cool, its use is not objectionable in magazines and confined spaces, and if any part is injured it can immediately be handled for repair.

5. There is no fear of explosion if a pipe or machine be struck by a shot.

6. On account of the high working pressure used, the machine can be kept small and light.

7. It is specially adapted for absorbing the energy of recoil of the guns.

Hydraulic pumping engine.—A constant supply of water at a pressure of 800 to 1,100 lbs. per square inch is maintained by large steam pumping engines, two of which are fitted in each battleship. A small auxiliary engine is fitted in addition for use when the machinery is being cleaned and repaired. A sectional elevation of one of the main pumping engines, showing the principal parts, is given in Fig. 853. This engine, which is of the horizontal tandem compound type, has two sets of steam cylinders and pumps secured to one bed-plate, and coupled to a crank-shaft with cranks at right angles to each other.

The H.P. cylinder is placed between the L.P. cylinder and the shaft. The L.P. piston has two piston-rods, which pass through chambers cast on the sides of the H.P. cylinder, and are secured to a crosshead to which the H.P. piston-rod and the pump ram are also secured. The slide-valves are arranged horizontally on the tops of the cylinders, and are both driven by one eccentric through a rocking shaft and levers. As the piston-rods and pump ram are connected to the same crosshead, most of the force of the steam is transmitted directly to the water in the pump. These pumping engines are not fitted with flywheels, and the resistance of the pumps being constant it is necessary to maintain the full steam pressure practically throughout the stroke. The point of cut-off is at ·95 of the stroke.

The pump piston has a sectional area equal to twice that of the ram, and is kept watertight in one direction by two L leathers.

The action of the pump is as follows.—During the out-stroke of the ram, water flows into the pump through the suction valve A, and fills the barrel. At the same time the water in the annular space B in front of the piston is driven out through the delivery valve C. While the ram is moving into the barrel during the return or in-stroke, the water in the cylinder is forced out through the intermediate valve D, half passing into the annular space around the ram, and the other half being forced through the delivery valve. The pump therefore delivers water during both strokes, but only draws water through the suction valve during one stroke. The delivery of water from the two pumps is very regular, and the arrangement has the advantage of keeping a constant pressure on the packing of the ram, which reduces leakage and prevents the entrance of air during the in-stroke. This is very important, as the presence of air in hydraulic machinery causes irregularity of working and may seriously increase the stresses on the parts.

As the water is used for working a considerable number of machines which are used intermittently, the demand on the pumps may be very irregular, and the speed of the engines must frequently and rapidly

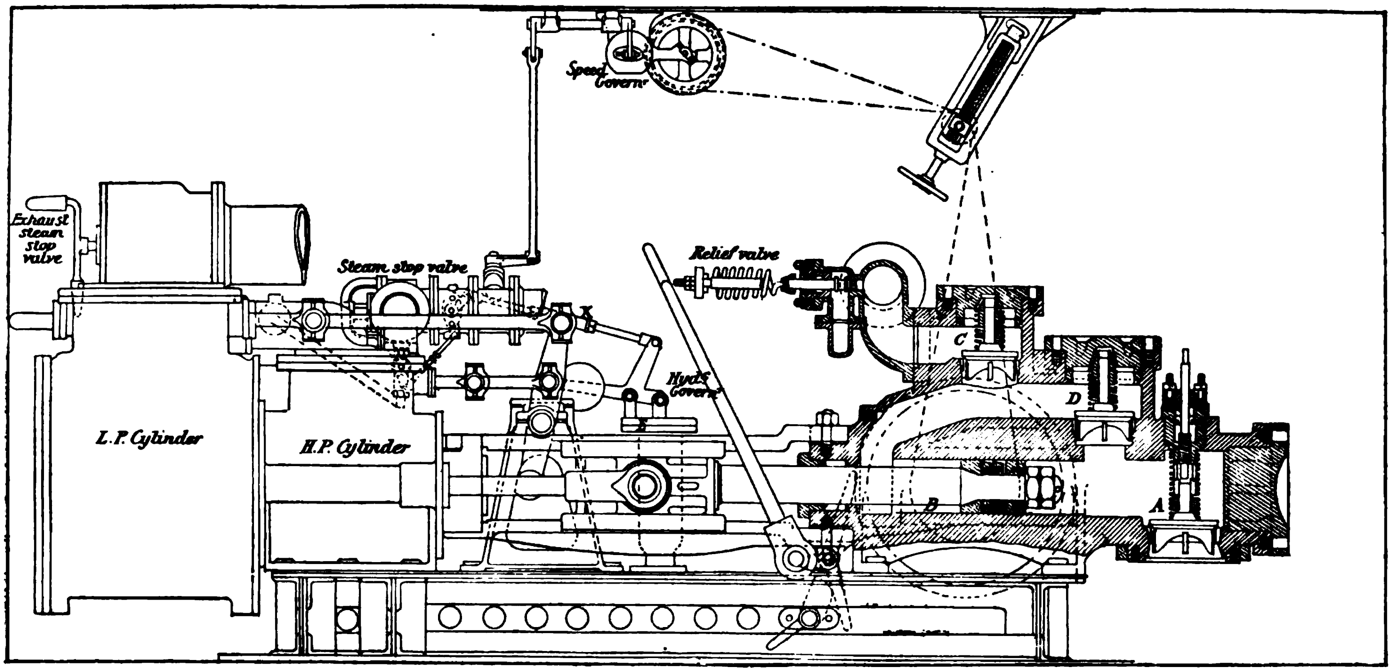

Speed Governr
Exhaust steam stop valve
Steam stop valve
Relief valve
Hydc Governr
C
D
B
A
L.P. Cylinder
H.P. Cylinder

Fig. 353.

change. If friction and the inertia of the moving parts were negligible, it would be necessary only to provide constant steam pressures on the cylinders high enough to balance the required pressure in the pumps, and the engine would automatically vary its speed so as to maintain a nearly uniform water pressure. Small pumps made on this principle are fairly satisfactory.

Hydraulic governor.—For large pumps, however, the variations of water pressure would be too great, and some means of regulating the steam pressure is required. The necessary variations of speed are too sudden and frequent to admit of hand control, and an automatic pressure governor is generally used. A section through one of these governors is shown in Fig. 354. A small hydraulic cylinder B is connected to the delivery pipe of the pumps. A plunger A which works in it is pressed down by springs, which are so proportioned that the plunger cannot rise until the water pressure reaches 950 lbs. per square inch and will have risen through its full travel when the pressure reaches 1,150 lbs. Leakage between the ram and cylinder is prevented by the cup leather C, shown unshaded in the figure. The upper end of the plunger is connected to a throttle valve in the steam pipe, and when the plunger is at the top of its stroke this valve is nearly closed, only allowing sufficient steam to pass to keep the engines moving slowly at 4 to 6 revolutions per minute, the water delivered from the pumps being returned to the suction tank through the relief valve shown in Fig. 353, loaded to about 1,100 lbs.

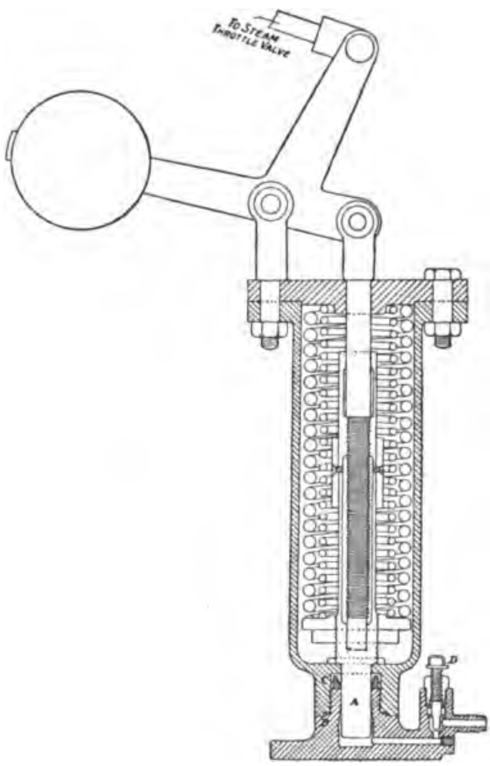


FIG. 354.

When one or more hydraulic machines are started, the pressure in the delivery pipe falls, and the springs force the plunger down and open the throttle valve until the speed of the engine increases sufficiently to bring the pressure back to the normal. To prevent a too sudden variation in speed a small valve D is fitted between the governor cylinder and the delivery pipe. This is regulated to only allow a slow motion

of the plunger. To prevent the dangerous racing which would result if a pipe burst or the suction tank became empty, and which would cause the plunger of the hydraulic governor to fall to the bottom of its cylinder and so admit full steam pressure to the engines, a centrifugal speed governor is fitted (see Fig. 353). The water used in the various machines is returned to the suction tanks. To reduce the friction of slide-valves, &c., and to prevent corrosion of internal steel parts, soft soap and mineral oil are mixed with the water.

Hydraulic gun-turning engines.—Hydraulic turning engines are used for rotating the gun turntables of turrets and barbettes. The turntables are carried on a number of flanged coned rollers, which move between accurately turned roller paths, one of which is secured to the ship and the other to the underside of the turntable. The rollers are kept at the correct distances apart by two circular rings of steel plate, which carry the axis pins of the rollers. The rollers and upper roller path are made of forged steel, and the lower path of cast-steel. Bronze racks are secured to the peripheries of the turntables. Pinions on vertical shafts gear with the racks, and the turning engines are connected to the shafts at the lower ends through suitable gearing.

The engines are of the three-cylinder oscillating type, and water is admitted and exhausted from each cylinder by a slide-valve which is moved by a crank pin on the cylinder trunnion. The valve-box also contains a reversing slide-block which by its movement starts, stops, reverses, and regulates the speed of the engine. As these blocks are large and the pressure high the friction between the working faces is great and they could not well be moved by hand. A small hydraulic reversing cylinder is therefore provided for this purpose. The piston-rod is connected to a weigh-shaft, which moves the three reversing slides simultaneously when water is admitted to one end or the other of the reversing cylinder by a small slide-valve, which is moved by handwheels fitted in the turntable close to the gun sights. The gearing is so arranged that the position of the reversing slide always corresponds to a definite position of the handwheel.

A brake is fitted on the engine shaft which comes into action automatically when the water pressure at the engine falls below a certain amount. This insures that the turntable shall always be under control. Duplicate engines and gearing are provided for each turntable. These hydraulic engines work very smoothly at any speed. They are quite free from any risk of dangerous racing, even if all load is removed, as the hydraulic resistances in the engine itself increase very rapidly as the speed rises.

In some ships the engines have been fitted in the turntables, the racks being secured to the structure which supports the lower roller path. In recent vessels care has been taken to arrange the parts of the turntable, and the guns and machinery fitted in it, so that the centre of gravity of the whole is situated on the axis of revolution.

In addition to working the turning engines, the hydraulic power is also used for working a number of hydraulic machines for loading and working the guns, raising ammunition, &c. All important machines, pipes, &c., are duplicated, and in the later ships each operation can be performed by hand if necessary, though at a much slower speed. In some ships electrical motors are fitted to perform a few of the opera-

tions of working the guns in case of break down of the hydraulic machinery.

Steam turret-turning engines.—The earliest turret ships were provided with steam turret-turning engines, the engine being controlled by a balanced differential valve similar to that described in Chapter XVII., which can be worked from the turret as well as at the engine. Steam-engines have also been fitted in H.M.S. 'Barfleur' and 'Centurion' for this purpose.

Air compressing machinery.—Machinery for compressing air for the purpose of charging and sometimes launching Whitehead torpedoes is now fitted in nearly all warships. The types of engines and pumps used for this purpose vary in design and arrangement, but in all of them the compression is done in two or more cylinders with successively decreasing diameters. The drawings, Figs. 355 and 356, of a two-stage compressor will serve to illustrate the principle. Fig. 356 is a vertical cross section through Fig. 355, on a larger scale, to indicate the detail more clearly. The pump has a cylinder A fitted in a casing B, through which latter, water is circulated to carry off the heat caused by the compression of the air. In this cylinder a combined piston and cylinder C, attached to the piston-rod of the engine, works. Air is drawn into the large cylinder on the down stroke through the annular valve D.

A little water and oil is admitted into the large cylinder during the suction stroke and, passing in as spray with the air, assists in carrying off the heat of compression. During the up stroke the air is compressed and forced through a number of holes past the valve E (shown unsectioned in the figure) into the moving cylinder. Only one of these small holes is shown. On the next down stroke more air enters the large cylinder through the inlet valve, and the air in the small cylinder is forced out through the delivery valve F. The air next passes through a coil of copper pipe, G, in the water casing, where it is cooled. The cooling water is circulated through the casing and also through the condenser H, in which the injection water is produced by condensation in a small steam coil, by the action of the moving cylinder which, in combination with suitable valves, one of which is shown at K, acts as a pump. Each cylinder of the compressor and also the casing is provided with a safety-valve.

The compressors are arranged in sets of one, two, or three, on one bed-plate. Each can compress 10 cubic feet of air to a pressure of 1,700 lbs. per square inch in 70 minutes. When working at their maximum capacity, the machines run at about 350 revolutions per minute. Special machines for torpedo boats and destroyers run at 500 revolutions per minute.

Separator column.—After leaving the cooling coils the air passes through a separator column L, where the injection water is removed. The separator column is formed of a steel tube with caps screwed on the ends. The top cap carries an inlet and outlet valve, and the lower one has a drain valve. A short length of pipe M is screwed to the inlet orifice, and extends down the inside of the tube for a portion of its length. The water and air on entry are therefore directed downwards, the water falling to the bottom and the air rising and passing out through the outlet valve to the reservoir. The drain valve requires to be opened frequently to blow out the accumulation of water. In a

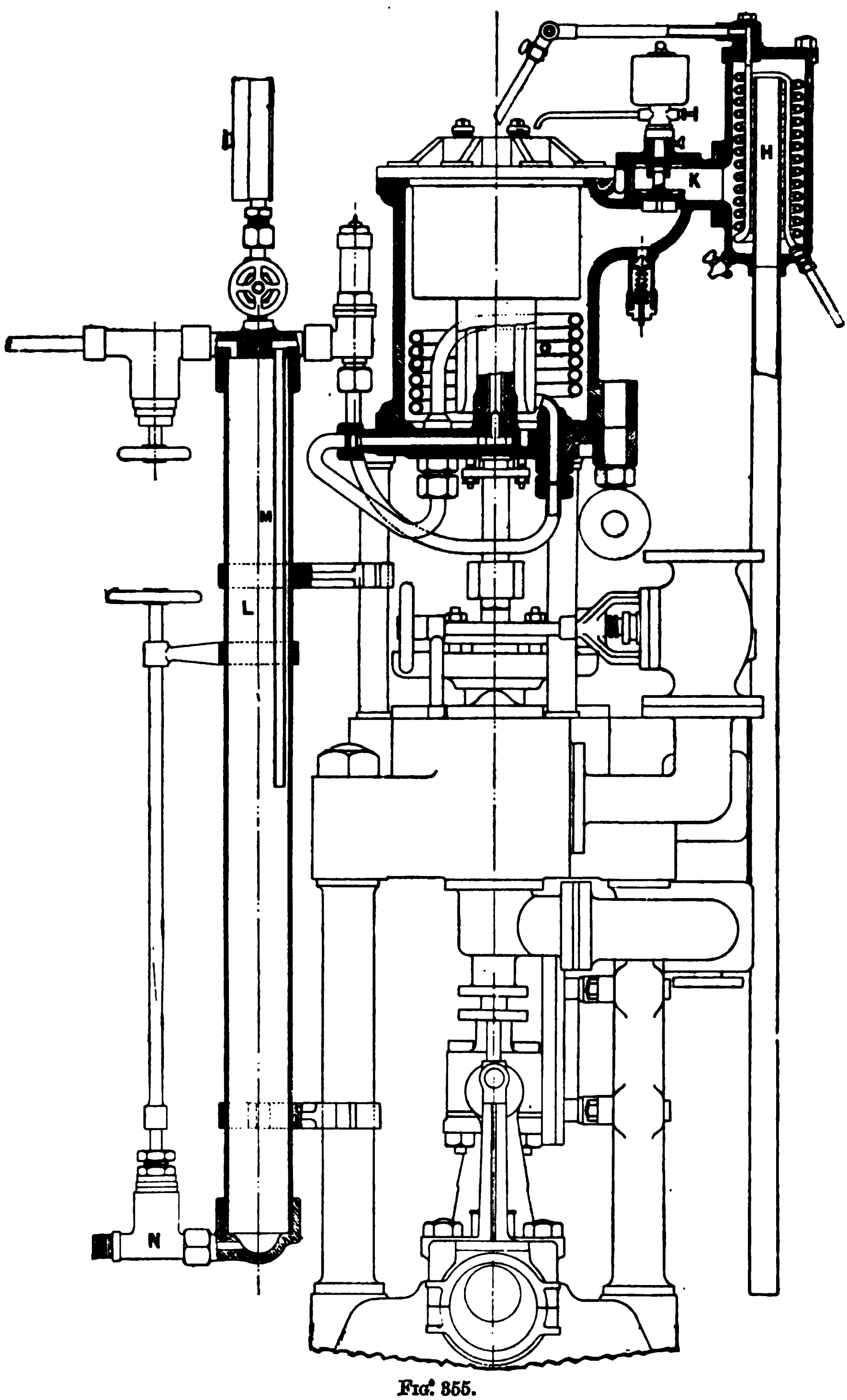


Fig. 355.

recent form the action is automatic, the valve being opened by the weight of the water which collects in a cup supported by a spring, and to which a balanced piston valve is connected.

Air reservoirs and pipes.—The air is stored in reservoirs of steel tubes arranged in groups of fifty. The tubes are fitted with gunmetal caps at the ends, and are connected together by screwed unions and connecting pipes. The tubes are about 3 inches in diameter, $\frac{3}{16}$-inch thick, and 6 feet long, and a reservoir of fifty tubes has a capacity of about $11\frac{1}{4}$ cubic feet. The whole of the air fittings are tested by a water pressure of 2,550 lbs. per square inch, the maximum working air pressure being 1,700 lbs. per square inch. The air is distributed to

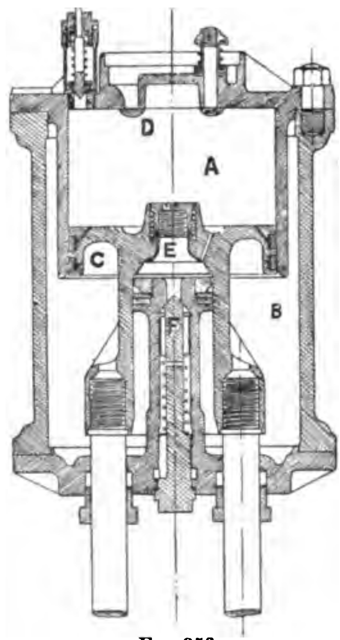


FIG. 356.

the torpedo tubes through copper pipes of $\frac{3}{8}$-inch internal diameter, the main leads being fitted in duplicate.

In all torpedo gear and fittings the greatest reliability and accuracy are necessary, as such very high pressures and sudden impulses have to be dealt with. Every precaution must be taken to prevent leakage, and too great an amount of care cannot be exercised in the fitting of every detail of the gear.

Boat-hoisting engines.—In most large ships, engines arranged to drive suitable winch barrels are now fitted for the purpose of placing the torpedo and other boats into the water, or of lifting them, and landing them in position on board in fixed crutches. Two engines and winch barrels are necessary, one for lifting the boats out of the water, and the other for 'topping' the derrick and bringing the boats inboard. These should be quite independent of each other. The derrick is

hinged on a swivel or ball joint supported by a bracket secured to the mast or signalling pole.

Ventilating engines.—In most vessels blowing engines are required for the purpose of providing a supply of fresh air for the crew spaces, as well as for the ventilation of the engine and boiler rooms. The engines are arranged to drive rotary fans, which draw air from ventilating shafts, and distribute it through the ship by ventilating trunks with openings in the several compartments. The outlets from these ventilating pipes are generally fitted with light gridiron valves,

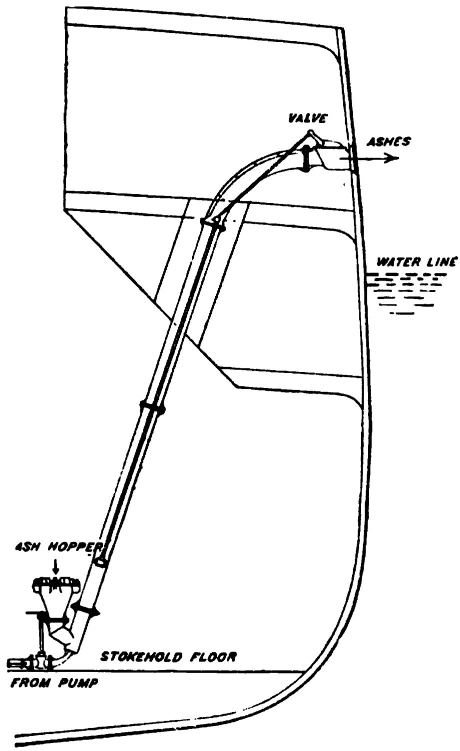


FIG. 357.

so that only such as may be required may be opened at any time. In modern warships, with closed stokeholds, blowing-fans and engines are also fitted for accelerating the draught in the boilers.

Ash-hoisting engines.—Small engines to drive winch barrels are fitted in most steamships for the purpose of lifting the filled ash buckets, &c., from the stokeholds, to enable the ashes to be thrown overboard. Some of these engines are fitted with reversing gears, so that the buckets are both raised and lowered by steam, or friction gear may be fitted to allow the barrel to be disconnected from the engines so that the bucket may descend by its own weight, the rate of descent being regulated by a brake. Overhead rails are fitted on the upper deck,

from the top of the ash-tube to the shoots at the ship's sides, to facilitate the discharge of the ashes, &c. A voice-pipe or gong is fitted for signalling between stokehold and deck.

Ash ejectors.—Various arrangements have been devised to obviate the necessity of raising ashes to the deck and thence discharging them. A steam ash ejector has been fitted in several vessels, but its low efficiency and the necessity of preserving fresh water renders such apparatus now inadmissible.

See's ash ejector is a more economical arrangement and is shown in Fig. 357. In this apparatus, which is fitted in many naval and other ships in which the raising of ashes on deck is objectionable, the ashes are placed in a trough leading to a pipe, a jet of water at a pressure of about 200 lbs. per square inch from one of the pumps is then admitted, and scours the ashes along the pipe into the sea. A small valve is fitted to permit the entry of air into the pipe during the discharge, which is above the water line.

Stone's underwater ash expeller.—As the result of experience in H.M.S. 'Africa,' Stone's underwater ash expellers are fitted in recent large warships in H.M. Service, and have also been fitted in some ships in the Mercantile Marine.

This apparatus discharges the ashes etc. below the water level, and they are projected clear of the hull, with its valve openings etc., by air pressure. It consists of a hopper containing a special design of crushing jaws, and immediately under this is a chamber communicating with a hollow single-ported revolving plug, into which the crushed material falls. From this drum the material is forced overboard, by air pressure, through a tube in the ship's bottom.

The machine is actuated by a steam engine which works the crushing jaws, revolves the hollow plug, and compresses the necessary air.

The material to be ejected is shovelled into the hopper and is crushed between the jaws, one of which is caused to reciprocate by a three-sided cam on the driving shaft. The shaft usually revolves at about 60 revolutions per minute, which gives 180 strokes per minute to the crushing jaws.

The crushed material then falls into the receptacle, below the hopper, and as the opening in the revolving drum comes to the top, the drum is filled. When the drum reaches such a position that the opening is connected to the discharge tube, the compressed air inlet in the casing is connected, by means of a port, with the inside of the drum, and the compressed air blows the ashes into the sea.

The drum in further rotating shuts off the connection to the sea, and at the same time cuts off the compressed air, and a little later, just before opening to the top, opens a small exhaust pipe, thus preventing the expanding air from violently blowing dust or ashes back into the hopper. The drum usually revolves at about 15 revolutions per minute.

When the apparatus is in use, the compressed air prevents sea water from entering, and a special form of sluice valve is fitted to the outlet tube for shutting off the sea when the apparatus is not in use.

This apparatus gives satisfaction when the inlet to the circulating pumps is not situate below a bilge keel, under which the ashes are also discharged. In such cases, portions of the discharged ashes may enter

the circulating inlet pipe and give trouble at the condenser, so that when it is inconvenient to shift the circulating inlet to the upper side of the bilge keel, and thus protect it from possible entry of ashes, a different arrangement is provided by which the ashes are discharged above the bilge keel as described below, and their entry into the condenser prevented.

Stone's patent hydro-pneumatic underline ash expeller.—In this arrangement a stream of water, which is drawn from the sea by

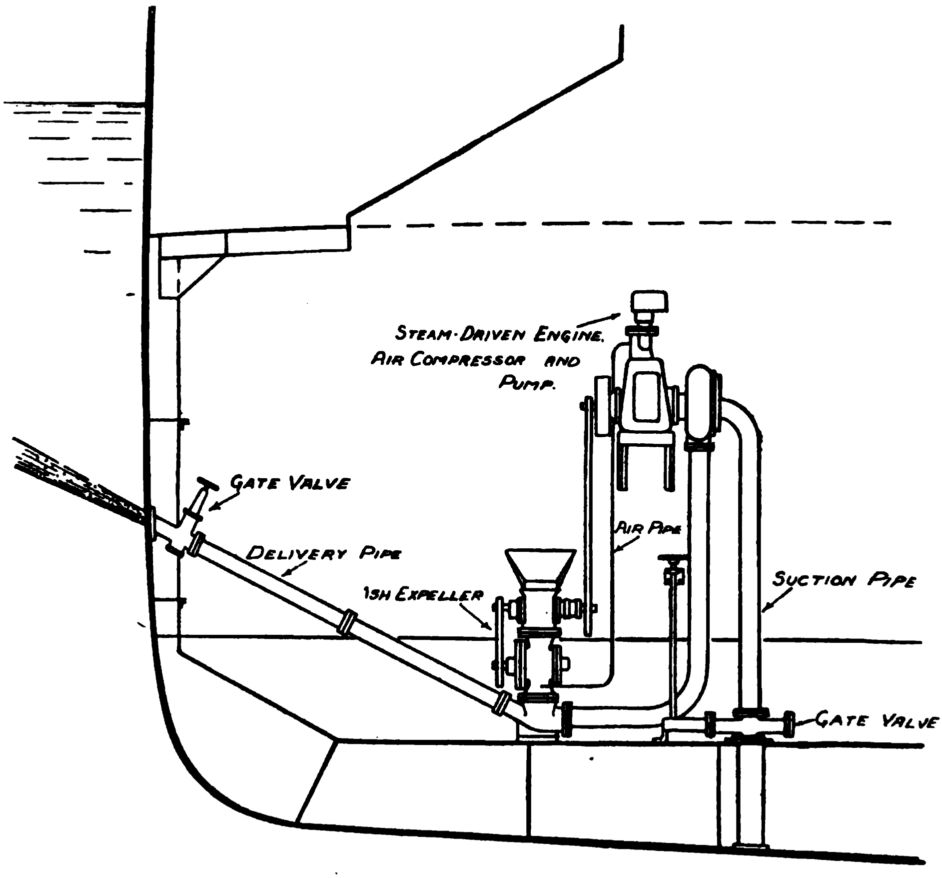


Fig. 358.

a centrifugal pump, is continually moving through a discharge pipe which passes underneath the expeller. The ashes and other refuse are shovelled into a crushing hopper, whence they pass into a double-ported revolving drum which alternately presents its opening to the hopper and to the moving stream of water beneath.

The ashes are then caught by the stream and carried along a discharge pipe, well clear of the ship's side and above the bilge keel,

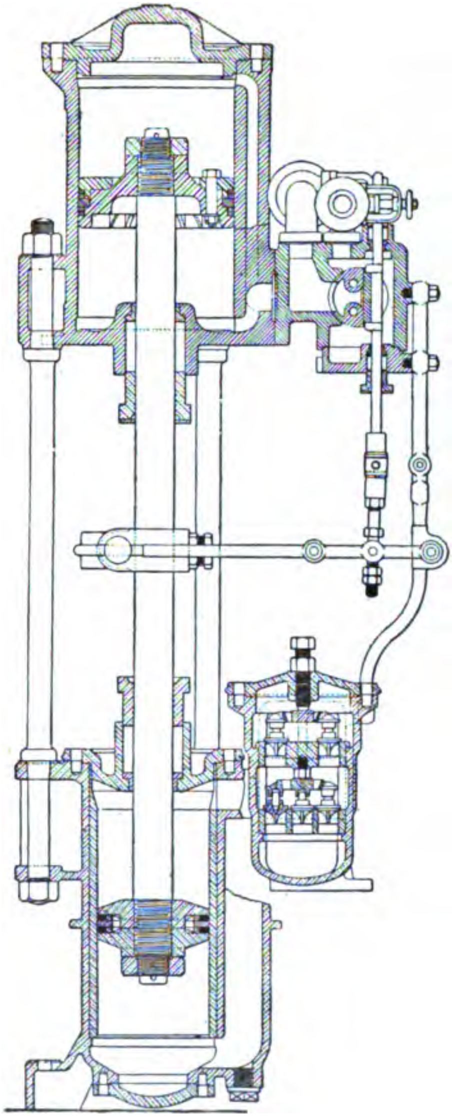
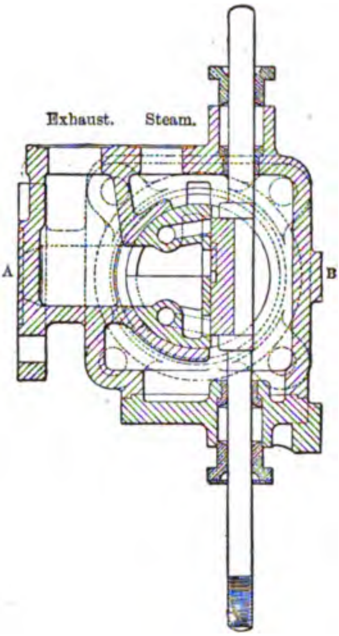


Fig. 359.



Vertical section.
Fig. 361.

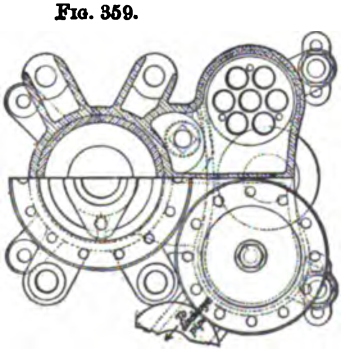
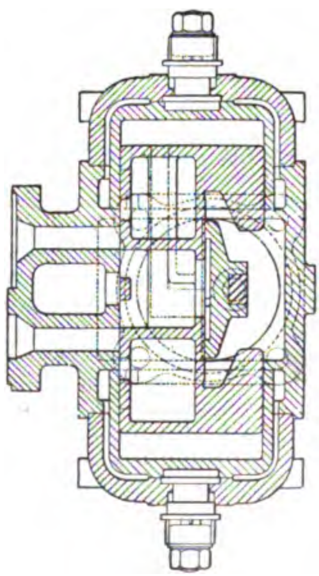


Fig. 360.


Horizontal section through A B
Fig. 362.

the main circulating pump inlet being below the bilge keel, which effectually prevents any of the discharged ashes from interfering with the main condenser pipes.

Sufficient compressed air is supplied to the drum to ensure against the ingress of water to the ship.

Both the air compressor and the centrifugal pump are coupled to a two-crank single cylinder steam-engine, which is also utilised for driving the crusher and the revolving drum.

Fig. 358 shows the arrangement of this apparatus.

Feed-pumps.—For delivering the feed-water from the feed-tanks to the boilers, pumps of a variety of designs have been employed. For many years, in the Royal Navy feed-pumping engines of similar type to the bilge-pumping engine of Fig. 340 were employed, except that the pump valves were made of metal instead of indiarubber, as in the bilge- and fire-pumps. These pumps have a constant stroke regulated by a crank shaft, and the valves are worked with eccentrics. They gave satisfaction for many years with steam of moderate pressures, even when they worked at high rates of speed. With the high steam pressures now used this design has not been so satisfactory, and it has recently given place to others, larger and slower in speed, and generally with slide-valves worked by a tappet action from the piston-rod either of its own engine, or of its fellow engine if of the 'duplex' variety. Fig. 340 will serve as an illustration of the crank-shaft variety of feed-pump, the valves, however, being altered to the metallic type. We will describe two varieties of well-known feed-pumps of the other type, viz. Weir's and Belleville's.

Weir's feed-pump (Figs. 359 to 362) is a vertical pump, double acting, with a set of inlet and discharge valves for each end of the pump arranged at the upper part of the barrel. The valves are a series of small ones milled out of solid metal and give a large area with a small lift. In each valve-seat for engines of say 1,500 to 1,600 I.H.P., there would be about seven of these valves in the suction part, and four in the discharge part. There is generally a single cylinder and single pump, with a separate liner fitted for the pump piston to work over, the pump piston packing being in the latest variety made of vulcanite. Sometimes solid pump pistons are fitted.

The steam valve arrangement is rather complicated and not generally understood, but in view of the very considerable number of these pumps in use in the Royal Navy and mercantile marine, and the importance of maintaining them in proper condition, we will explain the action in detail. It consists of a main valve for distributing steam to the cylinders, and an auxiliary valve for distributing steam to work the main valve. The main valve moves horizontally from side to side, being driven by steam admitted and exhausted from each end alternately; the auxiliary valve is actuated by a lever with fixed fulcrum worked by the rod of the pump. This auxiliary valve moves on a flat face on the back of the main valve as shown in Figs. 361 and 362, and in a direction at right angles to the latter.

Both the main and auxiliary valves are simply slide-valves, but the former is half round, the round side working on the correspondingly shaped cylinder port face, while the back of the valve is flat. Both ends of the main valve are lengthened so as to project beyond the port face

(see Fig. 362) and are turned cylindrical with flat ends. Caps are fitted on each of these ends, forming cylinders which are closed at the mouths by the flat ends of the main valve, which act as pistons, the length of stroke the piston can make being the full travel of the valve.

The auxiliary valve face has three ports (see Fig. 361), the centre one being the exhaust, and the two side ports being steam passages led through the piston ends of the main valve. The right-hand cylinder port passage is led through the left-hand end of the piston. the other passage being similarly led to the other end of the valve. These ports admit steam to the two small caps or cylinders at each end of the valve alternately, by which it is thrown from side to side.

Besides these ports, two other ports are formed on the auxiliary valve face leading to, and corresponding to, two ports on the half-round main valve face for admitting steam to the top and bottom of the cylinders. These ports on the auxiliary valve face are arranged to cut off steam before the end of the stroke and so reduce the speed, but the expansion chambers at each end of the main valve are fitted with bye-passes to admit steam for the full stroke when desired. This may be necessary, for instance, when starting the pumps, as then the cylinders may be full of water. These bye-passes are formed by notches cut in the edges of the caps, and may be opened or shut by turning the caps by means of the spindles at each side of the valve chest, and thus give a definite cut-off. There are separate bye-passes for up and down strokes, and the silent working of the pumps depends on the proper regulation of these bye-passes.

The auxiliary valve and face act just like an ordinary single-ported slide-valve, except that, besides the vertical movement of the auxiliary valve, the *valve face* moves across the auxiliary valve when the exhaust port is open to one of the chambers at the end of the main valve; the other chamber being in this position open to the steam, the valve is thrown over until the exhaust is cut off, which acts as a cushion. In this position the main and expansion ports are full open for the return of the piston, which moves at a rapid rate until the expansion is closed; then at three-quarters stroke the auxiliary valve closes the expansion ports, and the speed is reduced towards the end of the stroke, when the auxiliary valve opens the exhaust and throws the main slide for the return stroke.

Examinations necessary.—The steam valve must be overhauled as often as the main slide-valve of the propelling engine, as it works under far more severe conditions as to pressure, being between the boiler pressure and the condenser. It is not, as might appear from its shape, a piston-valve, but is an ordinary slide-valve, and as such must be kept tight by facing up in the usual way. It is most important that the curved ribs between the steam and exhaust ports should have a good bearing, and if they show signs of losing this, they must be filed and scraped in the usual way. There is practically no wear on the valve itself, so long as it is bearing properly on the bars; and it is only when it loses the bearing that it wears at all. If the high-pressure steam is allowed to leak past the bars, it speedily cuts them away, and hence the importance of the faces being regularly looked to and kept in proper order. A little attention from time to time will insure this. As the auxiliary valve holds the main valve up to its face, after bringing up the bearing

it may be necessary to put a liner between the valve spindle and back of valve to hold it up if it is slack. After lining up, the valve should be moved backward and forward by hand to see that it is not too tight. The face on the auxiliary valve must also be kept in good working order. The remaining parts require no special remark.

The Belleville feed-pump.—This pump is double acting, the steam cylinder having an ordinary flat slide-valve without lap, worked by the curved lever shown, which is moved at each end of the stroke by a projection on the pump-rod. A passage is provided at each end, so that steam is admitted uniformly all round the cylinder barrel, and not at the top only, which avoids bending forces on the rod. The steam pressure remains constant, therefore, till near the end of the stroke, when the projection strikes the valve lever and commences to close the steam

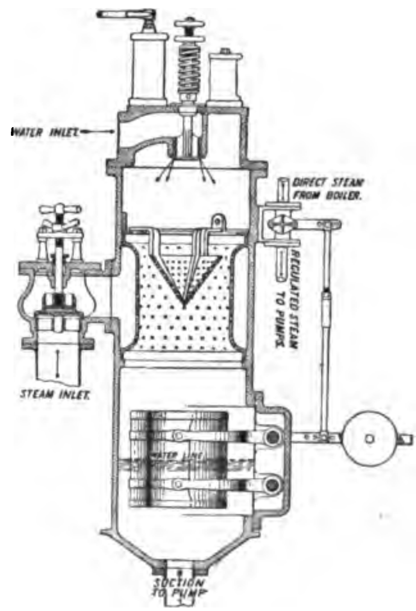


Fig. 364.

valve, so that the steam-pressure falls, and the motion would cease but for special fittings provided. Before the piston can commence the return stroke it is necessary that the valve should not only be closed but pushed sufficiently far over to reopen for steam on the other side.

To enable the steam already in the cylinder to complete the stroke and throw the valve over to the opposite side, an orifice is provided at each end of the pump-barrel, closed by levers and communicating with the suction chamber, so that when the pump piston nears the end of its stroke it strikes one of these levers and opens the orifice to the suction chamber, so that the pressure in the pump falls, and the steam in the cylinder, although cut off, is enabled by its expansive force to complete the stroke and reverse the steam valve, when the motion continues in the opposite direction.

The suction and discharge valves are a series of small ones, generally

eight in number at each end, four for suction and four for discharge. Small holes, about $\frac{1}{16}$-inch diameter, are made through the levers into the orifice leading to the suction chamber, so that a small quantity of water is always escaping from the pump-barrel, which causes the pump to keep slowly in motion even when the valves on the boilers are closed.

Feed-water heaters.—The advantage of supplying the boilers with feed-water approximating in temperature to that of the boiler has long been recognised, although the exact manner in which the practice should be conducive to economy has always been uncertain. Its beneficial effects as regards boiler preservation and reduction of racking stresses due to variations of pressure are well established. Even when the heating steam is taken from the boiler direct, so that theoretically there is neither a gain nor loss of heat by the process, large numbers of vessels in which such feed heaters are fitted report an appreciable gain in economy, and that when using the feed heaters, steam is more easily maintained than when they are not in use. When the heating steam is taken

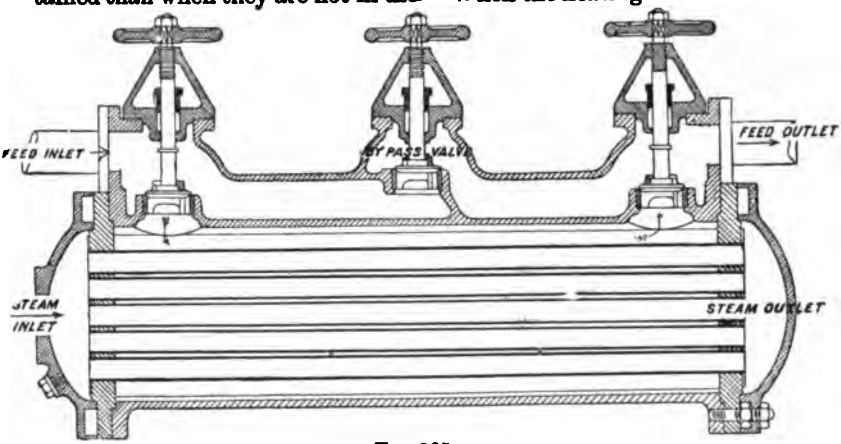


FIG. 365.

from the last receiver of the engine, or the exhaust steam of any engine, a gain in economy can be shown to exist theoretically. Feed-water heaters are now largely fitted in the mercantile marine, and results justify their adoption. They are also being fitted in some warships.

Weir's feed heater and regulator, Fig. 364, takes steam from the final receiver of the engine after it has done most of its work. It enters the heating chamber through a circular perforated ring, and there mixes with the cold feed-water, which is admitted through the spring loaded valve on the cover. The heated water falls to the bottom of the heater, whence it is removed by the feed-pump. A galvanised iron float is fitted to the bottom of the heater, which communicates by means of levers with the steam valve leading to the feed-pump, by which means the level is kept constant in the heater and the pumps are prevented from drawing air. The heat causes any air in the feed-water to be liberated, whence it can be drawn off by the cock on the top of the heater to the condenser or atmosphere.

Kirkaldy's feed heater.—In this apparatus, shown in Fig. 365, the steam does not mix with the feed-water, but the latter is conducted

through tubes, on the other side of which is the heating steam which is drawn from the boilers, or sometimes the exhaust steam from various auxiliary engines. It is therefore a surface-heater of similar construction to a surface-condenser, the tubes being rolled into tube plates in the ordinary manner. Bye-pass valves are fitted, so that when necessary the feed-water can be passed direct without traversing the heater.

Auxiliary condenser.—In the earlier ships in which the air-pumps were always worked by the main engines the exhaust steam from the auxiliary engines was led into the main-engine condensers in order to prevent loss of fresh water, a connection to the waste steam pipes being also fitted. The number of auxiliary engines in modern ships is so great that the unavoidable leakage into their cylinders was liable to affect the vacuum in the main-engine condensers, and when the main engines, and therefore the air-pumps, were not at work the auxiliary engines had to exhaust into the atmosphere, causing considerable waste of fresh water, unless other complications were introduced. To remedy this, separate auxiliary surface condensers are fitted in ships with air-pumps worked by the main engines, to receive the exhaust steam from all the auxiliary engines of the ship. These auxiliary condensers are fitted with independent circulating pumps, and with air-pumps that deliver into the hot-wells or feed-tanks, so that the condensed steam may be returned to the boilers by the feed-pumps.

They are of identical construction with those of the main condensers described in Chapter XX. Figs. 366 and 367 show details of the circulating pump, air-pump, and engine for an auxiliary condenser as commonly fitted in the Navy, metallic bucket, foot, and head valves being fitted. One engine serves for both pumps, the air-pump being worked below the crank shaft by double rods from the crosshead. Fig. 368 shows an auxiliary condenser with pumps and pipes.

The size and number of auxiliary engines being considerable in a modern warship, the auxiliary condensing power is also large. Except in small warships there are two auxiliary condensers, one in each engine room. In the case of large battleships the combined cooling surface is about 2,200 square feet.

In the most recent war ships, however, where the main condensers are fitted in duplicate on each side and independent air-pumps are fitted, auxiliary condensers are not being fitted, the auxiliary exhaust being led to the main condensers. The use of the closed exhaust described previously will prevent air leakage to condensers in this case.

Grease filters.—In Chapter X. the absolute necessity has been explained of preventing the entry of grease or oil in the boilers of vessels working with high-pressure steam. The best way to effect this desirable object is to limit as far as possible the use of oil for the internal parts of the engines. Many engines are capable of working efficiently without the direct admission of oil into the cylinders and slide-valves, the amount which finds its way inside owing to the necessary use of oil on the piston-rods and slide-rods being sufficient for internal lubrication. The machinery of torpedo boats and destroyers is found to work well under these circumstances; indeed, all oil cups are now omitted on such engines. The same method of treatment is now extended in many cases to larger engines with satisfactory results. Grease filters are a necessity in any case, as even the quantity of oil that enters through the piston-rods gradually becomes considerable.

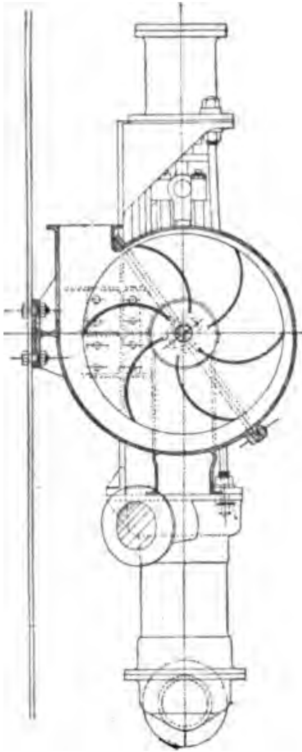


Fig. 366.

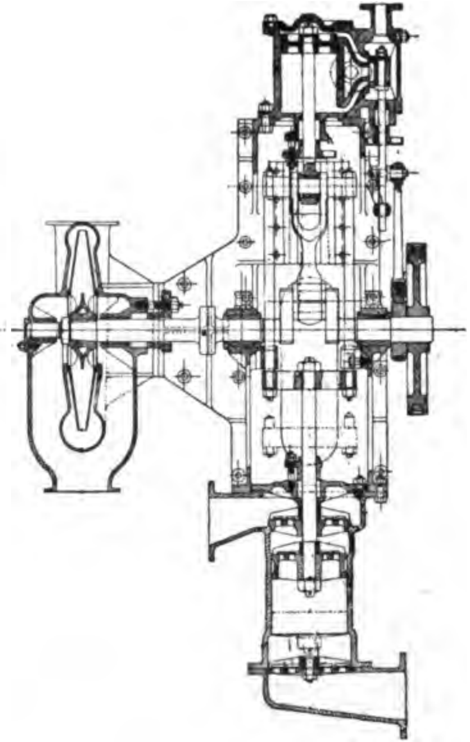


Fig. 367.

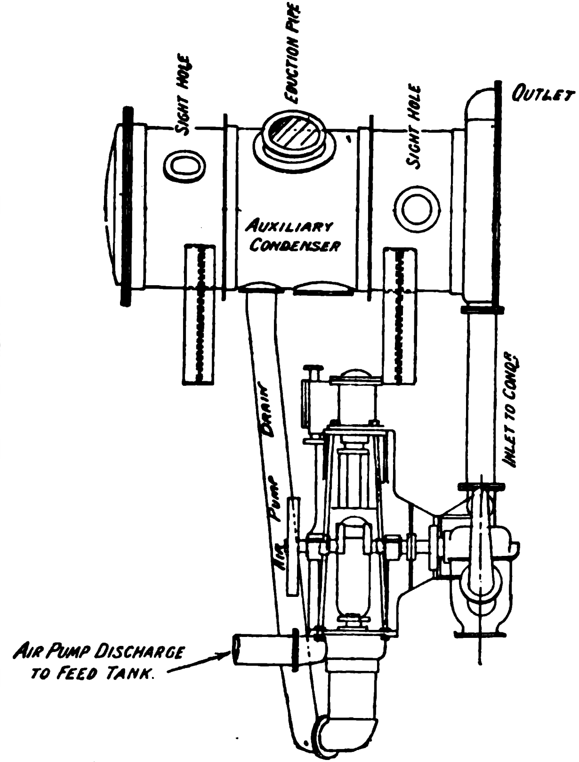


Fig. 368.

There are many types, but we select for illustration Harris's grease filter, fitted in the 'Campania,' H.M.S. 'Terrible,' and large numbers of other vessels. It consists of a considerable number of annular gratings threaded consecutively on a central spindle as shown in Fig. 369. On each of these gratings is fitted one or two sheets of filtering medium, consisting of towelling or flannel supported by wire gauze. These layers of filtering medium are shown at A A, in the sectional half of the diagram on the left. The gratings are so constructed that a large central space, B, is formed for the reception of feed-water, whence it enters to the spaces between the flannels through holes as shown, and after passing the flannels it makes its exit through similar holes in the circumference, shown in elevation on the right of the diagram, and proceeds to the outlet orifice. The course of the water is shown by

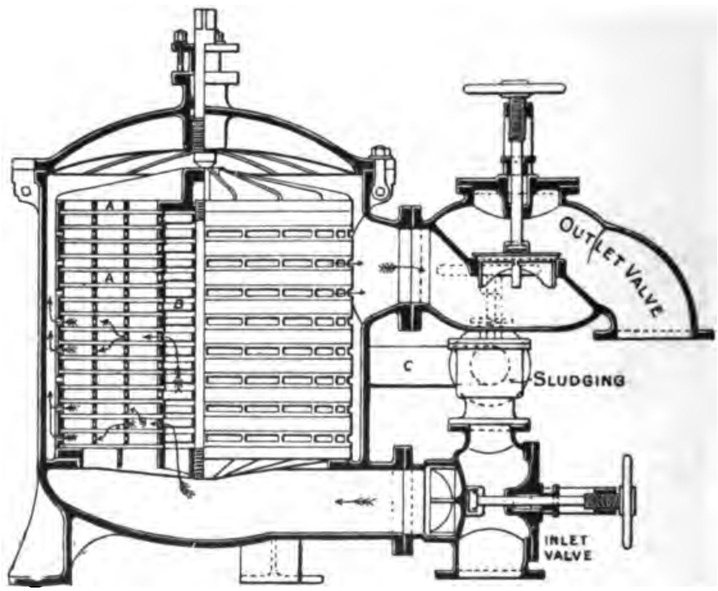


Fig. 369.

the arrows. This arrangement provides a large filtering area in a comparatively small space. When the cloths become dirty an arrangement of pipes, C, leading the inlet water to the reverse side of the cloths, and a steam cleaning jet, are provided, so that the grease can be washed off by a reverse current of water, a pipe and valve being supplied for carrying off the grease.

Governors.—The object of the governor is to maintain uniformity of motion of the engines when the resistance experienced is varied from any cause. In a marine engine considerable diminution of resistance may ensue in rough and stormy weather from the pitching motion of the ship when the propellers rise partly out of the water and increase the speed of the engines, causing what is technically called 'racing of the engines.'

The governor for a land engine usually consists of a pair of heavy balls rotated by the engine, which fly outward under the action of centrifugal force and close the throttle valve. This is not suitable for marine purposes, as the action of gravitation on the balls would be affected by the motion of the ship, so that the forces acting would become irregular, and other devices have therefore to be adopted.

The early marine governors were fitted to act directly on the throttle valve and required to be made large and heavy, as the motion produced on the valve was due entirely to the work accumulated in the flywheel. They therefore absorbed a considerable amount of power in working, and their action was not sufficiently rapid for modern engines. To decrease the weight and increase the sensibility of governors for marine engines, the more recent instruments of this class have been designed to cause the revolving or governing part of the apparatus to actuate a small valve only, for the purpose of admitting steam to a small governor cylinder, in which a piston connected to the throttle valve works. As the governing apparatus has only to work a small balanced valve, it may be made very light, so that the rapidity of action of the gear is by such means much increased.

Others still lighter and simpler are those in which, like the former, the governing apparatus is set in motion by the engine itself when the speed is increased, and in addition the force to close the valve comes also *from the main engine.* This is effected by arranging by various devices that the inertia of a moving weight causes an engagement with and closing of the thrcttle valve, by direct connection with some reciprocating part of the marine engine.

In each of the governors previously mentioned it will be noticed that an increased speed of the engine is required before the governor gear can operate, and they cannot anticipate and prevent such increase. This defect caused them originally to be of doubtful value, as it was combined with sluggishness in action, the throttle valve being generally closed only after the racing was over. The more modern instruments are, however, much more efficient, and are fitted in most large mercantile vessels making regular passages at high speeds.

Another entirely different class of governor is that which acts by variations of pressure at the stern of the vessel near the propeller, and not from engine speed variations. Racing being caused by diminished immersion of the propeller, it is accompanied by a diminution of pressure of water at that part which can be utilised to actuate the throttle valve. Such governors may therefore anticipate and prevent any increase of speed, and are more perfect in principle. Unlike those of the preceding class, however, they would have no effect in case of serious increase of speed due to such an accident as a broken shaft or propeller.

One of each of these types of modern governor will be described, commencing with the latter type.

Dunlop's governor.—This governor, shown in Fig. 370, consists of a sea-cock at the stern of the ship, opening into an air vessel or air chamber, A, so constructed that, by opening the sea-cock, water flows into the air vessel, and compresses the air contained therein to a pressure equivalent to the head of water outside the ship.

From the top of the air chamber a pipe B is led to the underside of an airtight elastic diaphragm, forming part of an apparatus in the

engine room. On the upper side of the diaphragm is a spiral spring, with means of adjusting its compression to balance the air pressure below the diaphragm. From the centre of the diaphragm a connection is made to the slide-valve of a small steam cylinder D, so constructed that its steam piston moves in exact accordance with the movements of the diaphragm. This steam piston is connected by suitable gear to the throttle valve of the engine whose speed is to be controlled.

The sea-cock being open, any variation of head of water outside

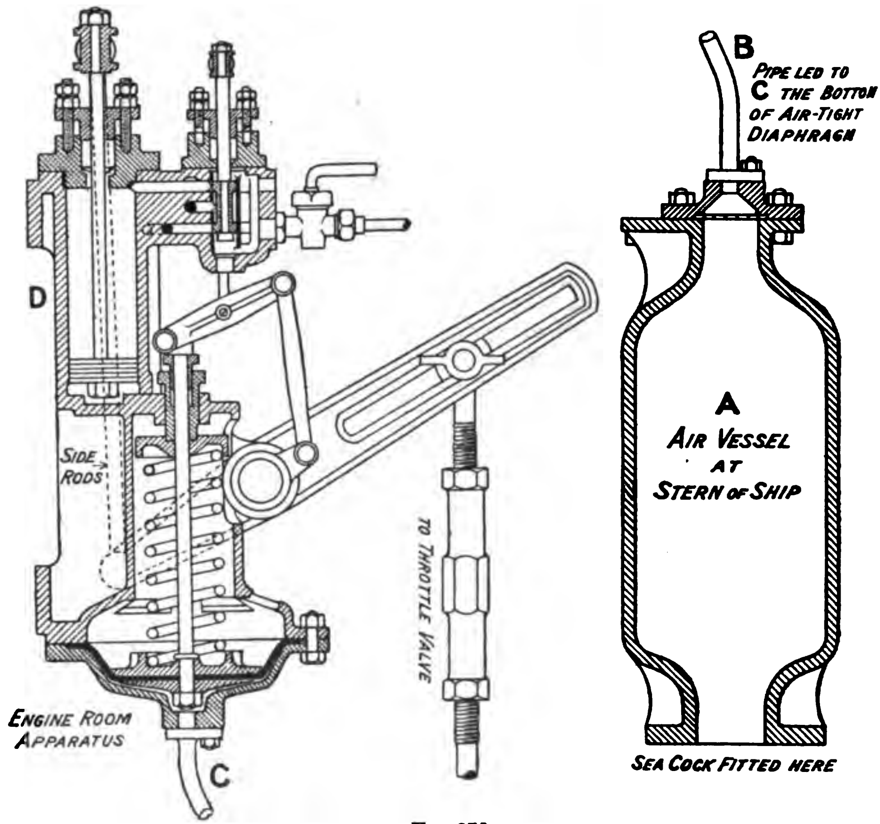


FIG. 370.

the ship is accompanied by an in-flow or out-flow of water through it, and consequently a variation in the pressure of the air contained in the air vessel, and also under the diaphragm of the engine-room apparatus, causing the diaphragm to move through such part of its travel as is requisite to enable the compression of spring and air pressure to balance one another again. Every movement of the diaphragm is followed by a corresponding movement of the governor steam piston, and consequently of the throttle valve of the engines under control, the time taken between the variation in the head of water

at the stern of the ship and the moving of the throttle valve being practically nothing.

The governor therefore anticipates any increase in the speed of the engines due to the propeller rising out of the water, and does not depend upon a variation in the speed of the engines to be controlled, before it acts. By adjusting the balance between the spring and the air pressure under the diaphragm, the diaphragm begins to fall and the throttle valve to close when the tips of the propeller blades rise to any desired distance from the surface of the water. The air vessel should be fitted as far aft in the screw tunnel as possible, the hole through the side of the vessel being placed about one-fourth the diameter of the propeller below the level of the centre of the shaft. The reports of the action of this governor in the mercantile marine are satisfactory. It is fitted in the 'Campania,' 'Paris,' and many other vessels.

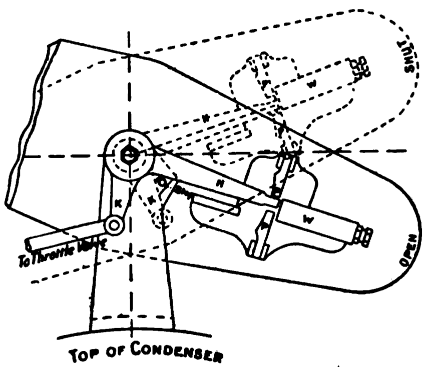


FIG. 371.

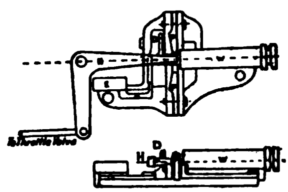


FIG. 372.

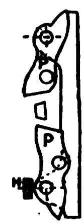


FIG. 373.

FIG. 374.

We come next to the other type, of which there are several varieties. One of them—viz. *Brown's emergency governor gear*—has already been described in dealing with reversing arrangements (see Chapter XVII.). Another example is the following, viz. :—

Aspinall's governor.—This governor consists of a hinged weight, W, operating two pawls, P, P, carried on a frame which is bolted to a pump lever, or other reciprocating part, as shown in Figs. 371 and 372. One of these pawls always projects, while the other lies close against the frame as shown on a larger scale in Figs. 373 and 374. The throttle-valve lever is so arranged that it is moved by these pawls when they project from the frame.

Under normal circumstances the lever H is in its lowest position and the throttle valve wide open, while the upper pawl projects and the lower one lies back as shown in Fig. 373, which represents the governor at the top of the stroke. The governor then travels freely past the throttle handle H without moving it. When the revolutions of the engines are increased by about five per cent. above the normal

speed, the weight, W, is left behind on the downward stroke, and therefore moves upwards relatively to the frame and is kept in that position by a detent arranged for this purpose. This movement of the weight reverses the position of the pawls, causing the bottom one to project and the upper one to be brought back. The bottom pawl then engages with lever H, lifting it throughout the whole upward stroke to the position shown by dotted lines in Fig. 371, and thus shutting off steam by closing throttle valve, the pawls being as shown in Fig. 374, which again represents the governor at the top of the stroke.

On the return stroke the detent is lifted by passing the lever H, liberating the weight W, which if the racing has stopped, falls, and the position of the two pawls is again altered, the top pawl now engaging with the lever H, depressing it, and thus reopening the throttle valve. If the racing has not stopped, the pawls remain as in Fig. 374, and the throttle valve remains closed. An emergency gear is also provided at E, the details not being shown in the figure, which only comes into operation in case of very excessive racing, such as on losing a propeller, or breaking a shaft, in which case an additional arm situated at A, is left behind and locks the weight W in the 'shut off' position, thus preventing the reopening of throttle valve.

Distilling apparatus.—The fresh water used on board ship for drinking, washing, culinary purposes, and for making up the waste of the feed-water for the boilers, &c., is produced from sea-water by the process of distillation. In addition to its convenience, there is no doubt that this practice has added greatly to the health of the Navy, as the water thus obtained is perfectly pure, which is impossible to insure in shore water in many ports of the world.

The distilling condensers first fitted were for drinking water only, and were simple condensers for the steam produced in the boilers, and any impurities in the boiler water often found their way by priming, &c., into the distiller. For this reason a separate boiler was usually fitted for distilling and auxiliary purposes, which was not fed with the greasy surface-condenser water, but with clean sea-water. This boiler was of such construction as to facilitate, as far as possible, the removal of the scale formed by the evaporation of the sea-water. The losses of feed-water from the boilers through leakage of glands, joints &c., were made up by the occasional admission of sea-water, generally by a cock on the condenser which connected the steam and water spaces.

As pressures of steam increased, this became inconvenient, as the admission of sea-water and formation of scale in the main boilers was objectionable, so that fresh feed-water to make up waste became imperative. Other arrangements were therefore made, and separate evaporators were fitted which produced vapour from sea-water by contact with tubes filled with steam, taken either from the boilers direct, or from the receivers of the engine. They are really small boilers, with heat obtained from steam passing through tubes, instead of, as in an ordinary boiler, by heat obtained directly from the combustion of coal. This vapour is conducted to the distilling condensers for production of fresh drinking water, and a portion to the main or auxiliary condenser for making up the deficiency of boiler feed-water. The scale is thus deposited in the evaporator, which is specially constructed to admit of its ready removal. This evaporator, with the usual distilling

condenser in connection, is spoken of as a 'double distiller,' as the resulting drinking water is obtained by the agency of two distillations of water, first in the boiler and secondly in the evaporator. The condensed primary steam is returned to the boiler.

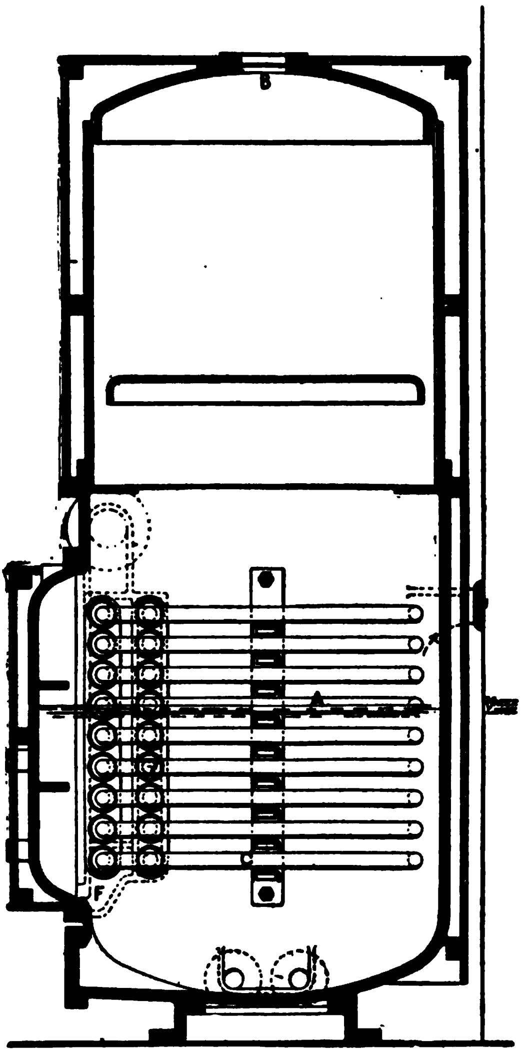


FIG. 375.

Low-pressure evaporators.—Remembering that evaporators are small boilers fed with sea-water, the considerations respecting temperature and density as affecting the deposition of scale explained in Chapter XX. become very important. Evaporators may be worked with steam of high pressure, when their size and weight for a given

production when clean, becomes small, but as the sulphate of lime scale is rapidly deposited on the tubes, their efficiency soon falls off and the scale is difficult to remove. The more recent warships of the British Navy are supplied with evaporators capable of doing the required evaporation with steam of low pressure, obtained usually from the closed auxiliary exhaust system (see page 308*b*). By this means, if the pressure of the vapour produced be kept at or near atmospheric pressure, and the density of the sea-water be kept below $\frac{3}{32}$, the deposition of sulphate scale may even be avoided altogether, and both increased efficiency and saving of labour results. There are many varieties of evaporators and condensers for marine purposes, but they all act on the same principle, and a description of one will be sufficient.

Weir's evaporator, Figs. 375 and 376, consists of a cylindrical shell with the evaporating tubes in the lower portion. Referring to the sketch, the heating surface is composed of coils (A) of solid drawn copper or brass placed horizontally. These tubes, with the exception of the bottom one, are secured at one end to a steam passage (D), whilst the other end is secured to a similar passage (E). The passage D can be connected to the auxiliary steam or the auxiliary exhaust system, and E is connected to chamber F by the bottom tube. F is in connection with the auxiliary condenser or feed-tank by the coil drain-valve. Immediately over the tubes are baffles by which the steam is separated from any water, thus preventing priming.

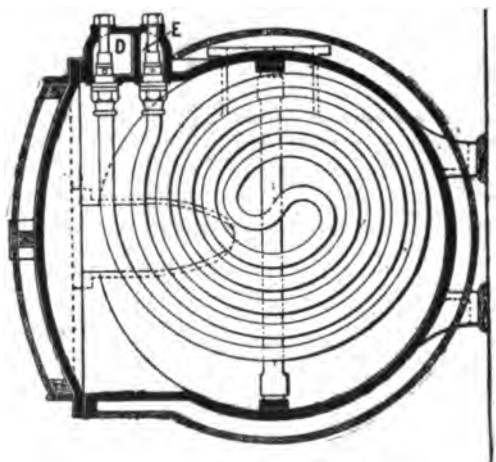


Fig. 376.

For feeding the evaporators with sea-water, and for pumping the brine away, a combined pump is generally fitted ; the feed-pump takes its suction either from the sea or from the distiller circulating water discharge, and the supply of feed-water to the evaporator is controlled by an automatic feed-valve worked by a float. Care is necessary in jointing up the tubes, but as these joints can be tested with steam before replacing the door, no trouble should be experienced from this.

The action of the apparatus is as follows :—The evaporator being filled to the working water-level, steam is admitted to D, from whence it passes through the evaporating tubes, giving up its heat to the sea-water on the outside of the tubes. From the tubes it passes partially condensed into E, and from thence through the bottom tube C to the chamber F ; from F it passes through the coil drain valve to the auxiliary condenser or feed-tank. The sea-water in the evaporator case is vaporised, and the steam generated passes through B on its way to the distiller, where it is condensed. The density of the sea-water

is kept down to the necessary amount by means of the brine pump, which pumps a certain proportion of the brine away. On its way to the pump, the brine is diluted by means of a connection to the evaporator feed pump sea suction, in order to reduce its temperature.

Distiller.—The distiller for condensing the secondary steam generated in the evaporator is of the same type as the ordinary surface condenser, with the exception that the tubes are generally placed vertically, and the ends of the tubes are expanded into the tube plates instead of being fitted with glands and packing. A combined pump is fitted

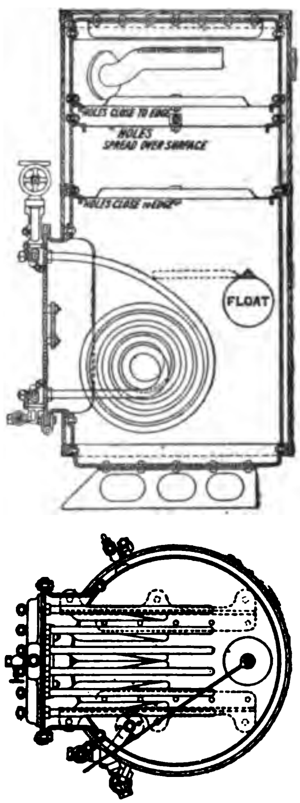


Fig. 377.

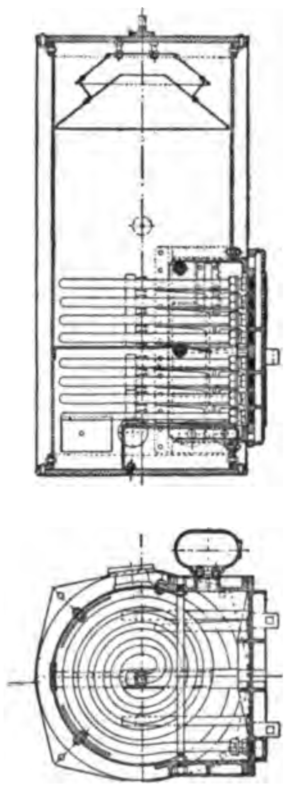


Fig. 378.

for circulating the cooling water through the distiller and pumping away the condensed fresh water. The distilled water is pumped by a steam pump into test tanks, and thence flows by gravity to tanks in the hold of the ship. The quality of the water in the test tank is occasionally tested.

In a large number of merchant steamers and ships of the Royal Navy, other kinds of fresh water making machinery are fitted.

Of these, Kirkaldy's and Caird & Rayner's evaporators are the most generally employed, and have each proved efficient.

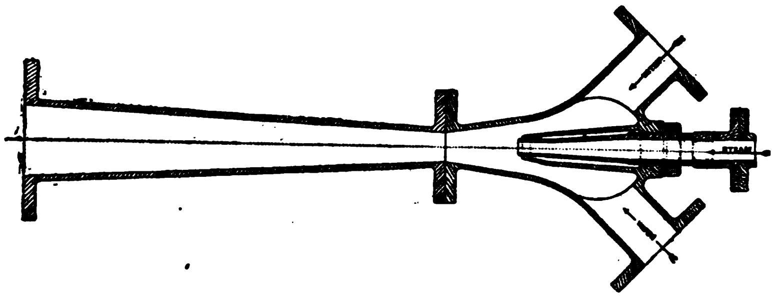


FIG. 379.

Kirkaldy's evaporator is largely fitted in the mercantile marine and Navy. It consists of a cylindrical shell with evaporating tubes in the lower part, but curved in a vertical plane in a different manner to Weir's, as indicated in Fig. 377. An automatic feed-valve and float is fitted to maintain a fixed water level.

Caird & Rayner's evaporator is also similar, the tubes being curved horizontally in the manner shown in Fig. 378.

The evaporating and distilling apparatus of these three firms, besides being fitted in large ships, are extensively fitted in torpedo boat destroyers and torpedo boats. A specially light form of apparatus is supplied for this purpose.

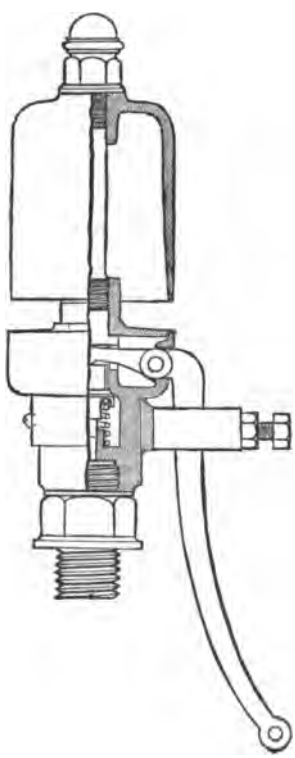


FIG. 380.

Steam injectors for filling fresh water tanks.—As explained in Chapter XX, the fresh water produced by the evaporators is stored for boiler purposes in the reserve fresh-water tanks. These tanks are filled with fresh water from the shore if convenient opportunities present themselves, generally by means of water boats alongside the vessel. The fresh water is generally drawn from the water boats and discharged into the engine feed tanks by means of a steam injector, a sketch of which is shown in Fig. 379. The centre orifice is supplied with steam, and water is allowed to flow in from either side of the ship and be discharged by the steam jet into the tanks. From the engine feed tanks the water overflows into the reserve fresh-water tanks, where it remains until required for the boilers.

Steam fog-whistle.—All steamships are fitted with steam whistles or sirens for signalling purposes, and for indicating the position of the

ship in case of fog, &c. A sketch of a steam whistle for use with moderate steam pressures is shown in Fig. 380. The sound is produced by the vibrations caused by steam issuing from the narrow annular orifice against the thin edge of the bell of the whistle. The bell is screwed on to enable its distance from the steam orifice to be adjusted to suit the pressure of steam used. The steam pressure acts on the top of the valve and tends to keep it closed, as also does a spiral spring. The valve is opened by a lever, and string or wire, led to some convenient place for working it.

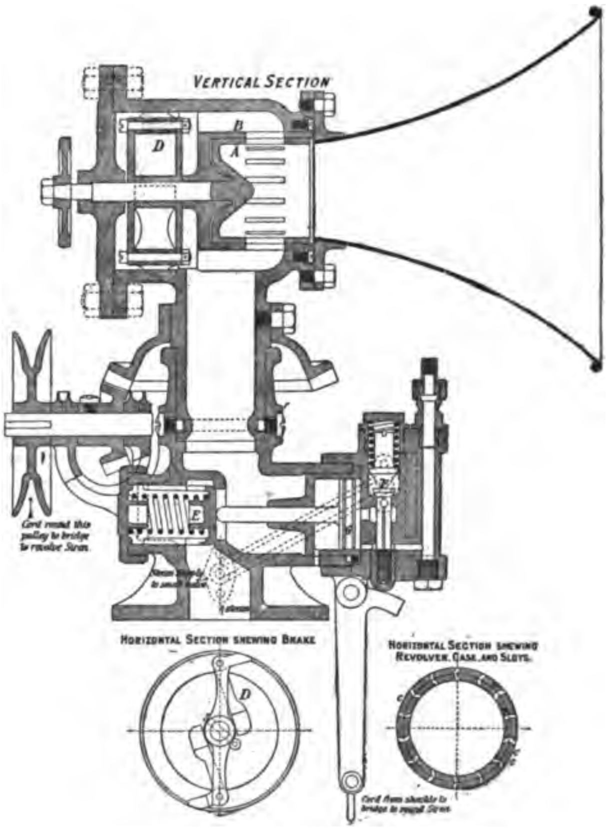


FIG. 381.

The siren is another and much more powerful instrument for signalling purposes at sea. In Holmes & Ingrey's siren, adopted in the Royal Navy and illustrated in Fig. 381, the steam passes through a number of narrow slits on the surfaces of two cylinders with horizontal axes, the inner one of which, A, revolves within the other, B, which is fixed, and the steam then issues to the atmosphere through a trumpet-mouthed orifice to increase the volume of sound. To assist in starting the siren, should the main slits not be opposite one another,

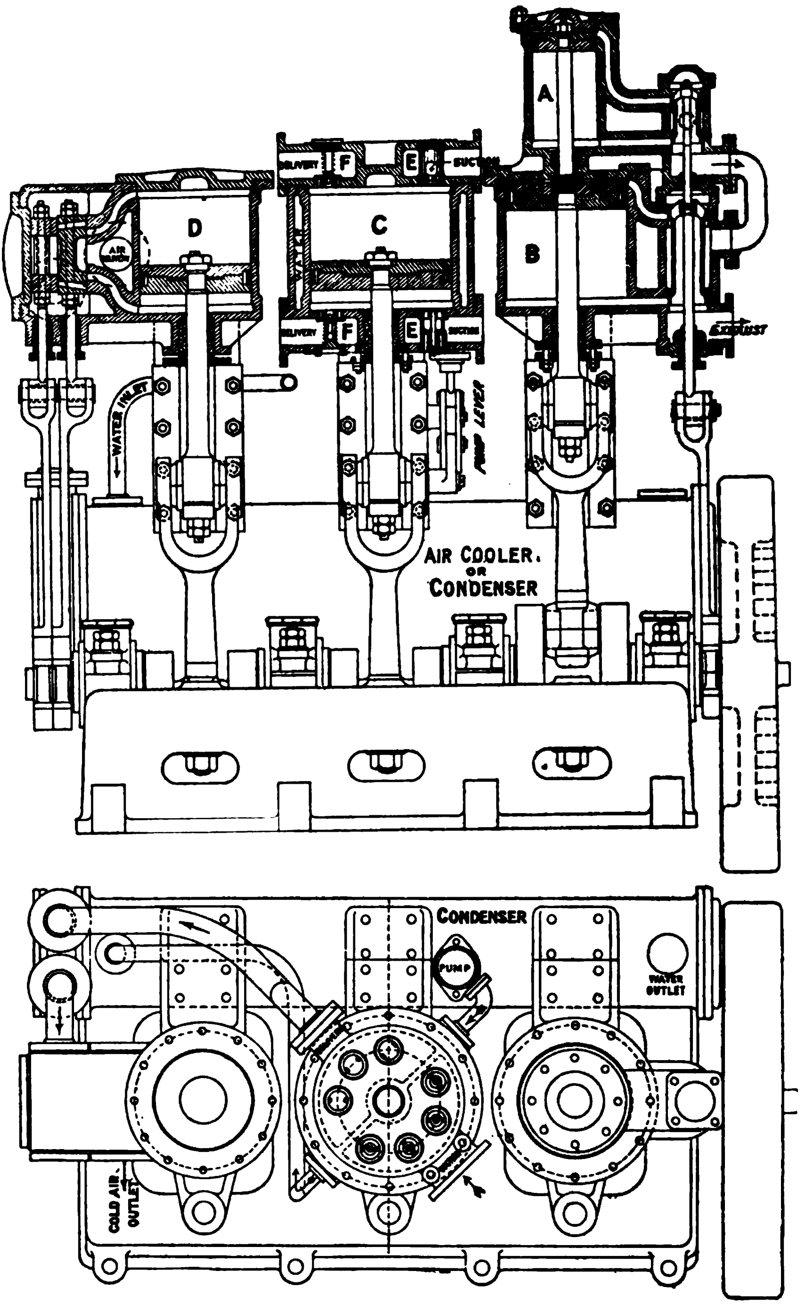

A
B
C
D
DELIVERY
F
E
SUCTION
DELIVERY
F
E
SUCTION
WATER INLET
PUMP LEVER
EXHAUST
AIR COOLER.
OR
CONDENSER
CONDENSER
PUMP
WATER OUTLET
COLD AIR OUTLET

Fig. 382.

heat in the air. This heated air, passing through the tubes of the air cooler, is cooled by the circulating water, and is then led to the valve-chamber of the expanding cylinder D. The compressing cylinder is fitted with a water-jacket through which the circulating water passes on its way from the air-cooler overboard. In the cylinder D the cut-off valves are so arranged that the proper quantity of air is admitted before supply is cut-off, and during the remainder of the stroke, the air expands, and therefore does work on the piston, heat being expended in the process in exactly the converse manner to the generation of heat in the compressing cylinder. As the air has been deprived of its surplus heat in the air cooler, the equivalent of the work it does is absorbed from itself, and a considerable lowering of its temperature results.

Snow box.—This cold air is then exhausted through the exhaust orifice of the slide valves and led first to the 'snow box' (a small accessible chamber in which the snow formed from the moisture is deposited), and thence to the cold chamber in which is the supply of meat or provisions, and where it displaces air of higher temperature.

Refrigerating Chamber.—The refrigerating chamber is insulated by lagging its bulkheads, ceilings and floor, with silicate cotton or other non-conductor, a teak lining being fitted over this to form the inside surface. The chamber is generally divided into two parts, a cold meat storage room and a vegetable room, as different temperatures are required to properly preserve the different provisions.

Ammonia Anhydride System.—This system, on account of its lightness of construction and high efficiency as compared with the cold air system, is largely used in the mercantile marine for refrigerating rooms. The poisonous nature of ammonia gas, however, precludes its use in the Royal Navy for this purpose owing to the position of the refrigerating chamber in the ship, but machines on this system are now largely fitted for ice-making purposes, the machines being placed on the upper deck, where little danger is to be apprehended in the case of a considerable escape of gas; a drenching arrangement is however fitted to all ammonia connections, by means of which any leakage of ammonia can be readily absorbed.

The machine consists of four essential parts, viz., A, the compressor, B, the condenser, C, the expansion valve, and D, the refrigerator (Fig. 382A). The compressor consists of a cylinder the piston of which is connected to a crankshaft revolved either by a steam-engine or by an electric motor: this compressor is single-acting, having head valves and a valve in the piston. The gland and stuffing-box are of the ordinary type, but packed with a special packing, and also kept tight by a seal of mineral oil pumped in by a hand-pump. The ammonia gas is first compressed in A; it then passes into the receiver, where any oil or dirt is separated from the ammonia; from the receiver it passes into the spiral coil in condenser B where the latent heat given up on changing from a gas into a liquid is removed by the circulating water pumped outside the coil through the condenser. The liquid is now allowed to expand or boil by passing through the expansion or regulating valve C into the refrigerating coils D, which are surrounded by brine, and this brine supplies the heat necessary to boil or vaporise the liquid, the gas being all the time sucked away, as it is given off, to be again compressed into a liquid in compressor A, and the cycle

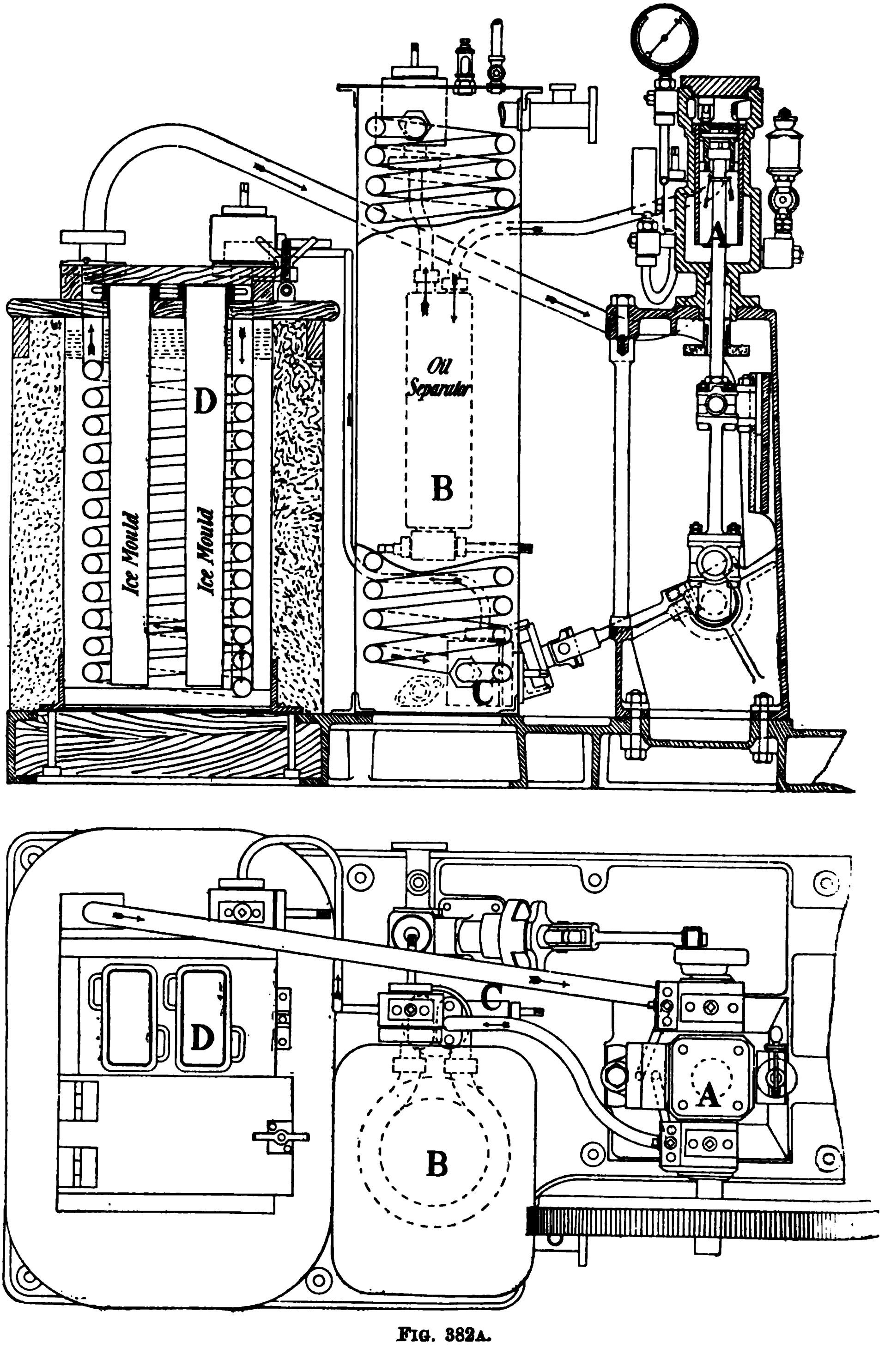


FIG. 382A.

repeated. In the brine are placed ice moulds, in which the water to be converted into ice is placed.

When used for refrigerating purposes the same system is adopted, and the cooled brine circulated through a series of pipes placed on the roof and sides of the cold chamber. On account of the action of ammonia on brass or copper, neither of these metals can be used, and all parts including gauges are made of steel or wrought-iron.

Carbonic Anhydride System.—This system is worked on the same principle as ammonia machines, but using carbonic acid gas (Fig. 383). These machines are more efficient than the dry air type, but slightly less so than ammonia machines, in addition to which the critical temperature of carbonic acid gas is 88° F., and it is impossible to liquefy CO_2 by compression in a climate where the sea-water is above 88° F.

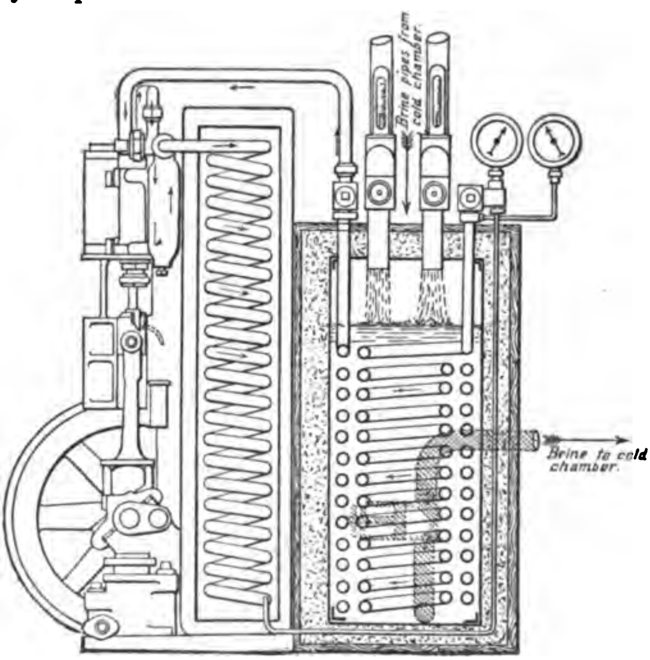


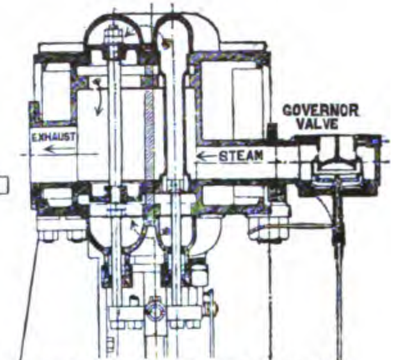
FIG. 383.

The necessary working pressure in the CO_2 machine is also much higher than that in the ammonia machine, being about 1,000 lbs. per square inch in the former and 200 lbs. in the latter.

It however has the advantage over the ammonia system that, provided the compartment in which the machine is placed is well ventilated, it can be placed below deck without danger. Machines on this principle are largely used in the mercantile marine, but in the Navy their use has been confined, principally due to the weight of stores necessary, to a few ice-making machines and magazine cooling plants.

Electrical machinery.—The dynamos used on board ships for internal illumination, and for working search lights and motors, are of the direct driven type, and are shown generally in Figs. 384 and 385.

Fig. 385.



SECTION THROUGH AB SHOWING VALVES

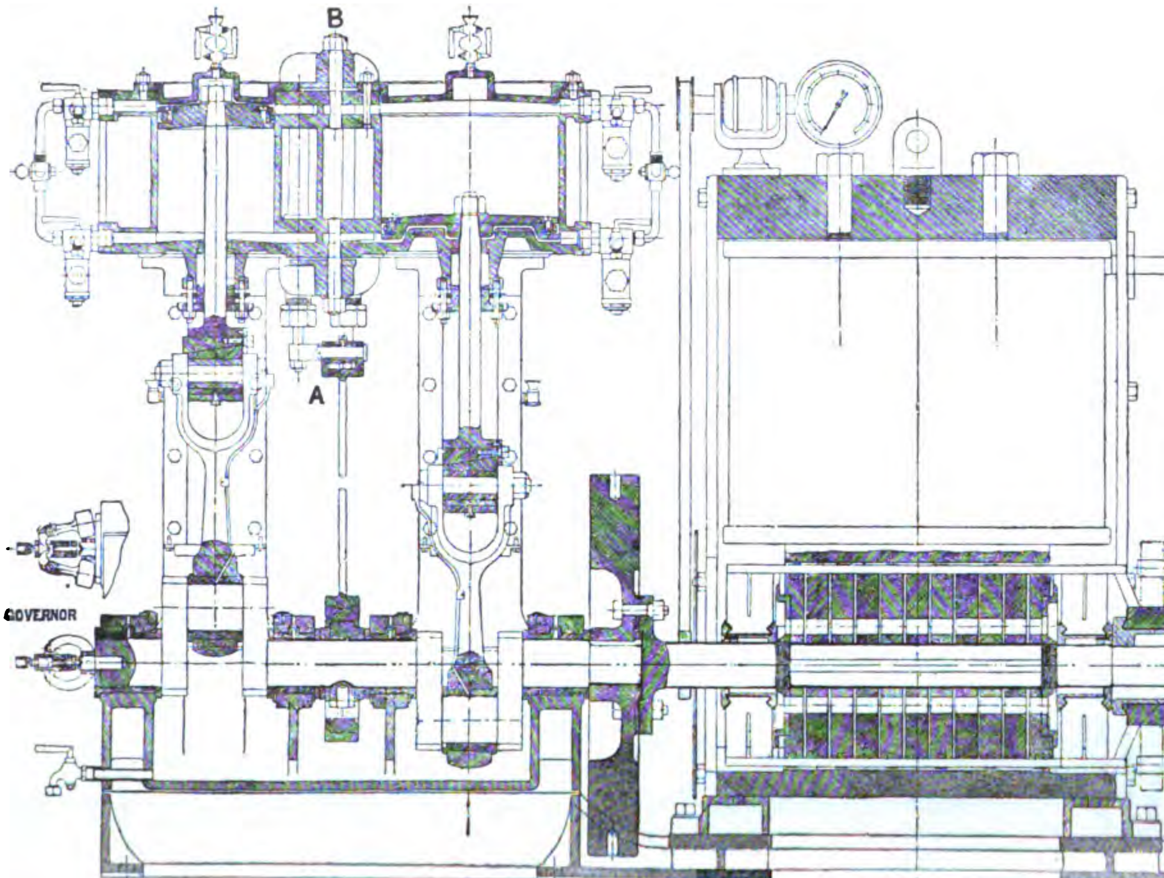


Fig. 384.

Each set of machines usually consists of a compound-wound dynamo with its armature shaft coupled to the crank shaft of a vertical double acting open-type steam-engine. The engines have two cylinders side by side, and the two cranks are 180 degrees apart. Most of the engines are compound, but some simple engines are in use in ships where the steam pressure is low. The pistons and other reciprocating parts attached to each crank are carefully balanced, and a heavy fly-wheel is fitted on the engine shaft, at the dynamo end, which conduces to steady running. A governor is fitted on the shaft and operates a balanced throttle valve. In Fig. 385, the two valves are worked by the same eccentric, the H.P. top port being open to steam by the amount of the lead, the exhaust from the H.P. proceeding as shown by the dotted arrow through the lower chamber to the bottom port of the L.P., while the upper port of the L.P. is open to exhaust.

Type of dynamos.—The dynamos are usually of the inverted two-pole type, and are carried on an extension of the engine bed. They have drum armatures and the field magnets are compound-wound to give a constant pressure of 80 or 100 volts for any current from zero to the maximum, while the speed is maintained constant. The usual speed is 320 revolutions per minute. The machines used in the Royal Navy are proportioned for maximum currents of 100, 200, 400 or 600 ampères, and generally three dynamos are fitted in each ship. New ships for the Royal Navy are provided with a pressure of 100 volts in order to reduce loss in the cables between the dynamos and lights. This loss is proportional to the square of the current flowing in the cables, and with a higher voltage a proportionately less quantity of current is required for the same amount of light.

All the machines are connected to a switch-board in a central position, from which the current is distributed to the various circuits for lighting, motors, &c. This board is so arranged that a circuit can be quickly changed from one machine to another, but no circuit can receive current from two machines at the same time. The most recently fitted dynamos are of the iron-clad type, the field magnet coils and the armature being almost entirely surrounded by iron to reduce to a minimum the leakage of magnetic lines of force which may affect compasses or chronometers in the neighbourhood. An illustration of this type is given in Fig. 384.

Construction of armatures.—The armature core of the dynamo is built up of thin discs of soft iron slipped over metal sleeves which are keyed on the shaft. The discs are insulated from each other by thin sheets of asbestos paper to prevent loss of energy and heating due to eddy currents, and are kept in place by clamping plates and end nuts.

The conductors on the armature which carry the current are made up of copper wires twisted together and pressed to a rectangular section. They are insulated by a covering of varnished tape. Usually two lengths of bars are used. They are placed round the periphery of the armature longitudinally, long and short bars alternating, their ends overhanging the core; all the ends at one end of the armature projecting the same distance. Projections are fitted into the core at intervals which drive the conductor bars. These projections are insulated by mica slips. The bars are kept in place by bands of steel or bronze binding wire tightly wound on and soldered. Mica strips are placed under the bands to prevent injury to the insulation of the bars.

Each bar is connected at each end by bent copper strips to another bar almost diametrically opposite to it, so that the whole of the bars and end connections form one closed circuit. The projecting end of each long bar is also connected to the nearest commutator segment, the number of segments being equal to the number of long bars. Two or more pairs of brushes bear on the commutator to collect the current, so that any brush may be lifted off without interrupting the circuit.

Field magnet windings.—The field magnet winding consists of shunt and series coils, wound on a frame which fits over the upper pole piece. The shunt coils are of small wire and are of high resistance. The ends of the wire are connected to the machine terminals. The greater part of the magnetisation is due to these coils, so that at full speed, and when no current is being taken from the machine, the electric pressure is normal, i.e. 80 volts, or in latest ships 100.

The series coils are formed of thick copper bars and convey the whole current generated. They provide additional magnetisation proportional to the current flowing in them, and so compensate for the additional pressure required to force this current through the machine. By the combination of the two sets of coils the pressure is thus independent of the current, so long as the speed is constant.

In the largest machines there are two distinct armature windings laid on side by side, the bars of the two windings alternating as also do their respective commutator segments. The two windings are connected in parallel by the brushes which all have a bearing rather wider than the angular width of two commutator segments.

Commutator.—To obtain satisfactory working the commutator must be kept true, and should be frequently examined to see if flats are being formed on it. These are due to unequal wear of the segments, generally from sparking, and they rapidly increase if neglected. If slight, they may be removed by filing while the machine is running slowly with the brushes off. If considerable, the commutator must be turned up. This is generally done in place, using a lathe slide rest clamped to the bed plate; the engines being run as slowly as possible. After turning the commutator should be polished.

Brushes.—The brushes must be carefully filed to fit the commutator curve, and to bear evenly from side to side. The most satisfactory way is to make a pair of hard wood or metal clamps, to hold the brush while being filed, having their ends of the correct shape. The brushes must be carefully set in the holders with all the tips of each set in a line, and the tips of the two sets bearing simultaneously on diametrically opposite commutator segments. Generally two segments are marked at their ends with crosses to assist in this adjustment. The electrical parts of the machine rarely require attention but should be occasionally tested to see that the insulation of the circuits is good.

Adjustment of bearings.—The engines should be very carefully adjusted to obtain quiet running. The centre line of the shafts should be maintained at its original height, which is usually slightly above the axis of the bore of the pole pieces. If so adjusted part of the weight of the armature is borne by a vertical force due to the uneven distribution of the magnetic flux of the field magnets, and this relieves the bearings. A gauge, by which the distance of the tops of the shaft journals below the planed parts of the bed can be readily measured

when the caps and upper brasses are removed, is useful for this purpose.

To insure steady running a heavy fly-wheel and good governor are necessary, and the latter should be kept in good condition and adjustment. When the load on the machine is gradually taken off the increase of speed at no load should not exceed 5 per cent., and when suddenly removed the temporary increase should not exceed 10 per cent.

The governor.—The governor should be frequently examined to insure its parts working freely without backlash, as friction causes the objectionable defect known as 'hunting.' This is a defect to which sensitive and not very stable governors are often subject.

If the governor gear is stiff and a reduction of load on the engine occurs, the governor does not act until the engine has increased its speed sufficiently for the increased centrifugal force to overcome the spring and the frictional resistances. When this speed is reached the valve will commence to close, but as the centrifugal force and the resistance of the spring are nearly equal for all positions of the valve, and because the friction of the parts in motion is less than their friction in repose, the valve will continue to close until the energy of motion of the gear has been absorbed by the frictional resistances and the work done in overcoming the unbalanced part of the spring's force. The resulting valve opening will be too small for the normal speed and the engine will be slowed too much. The opposite action will now commence, resulting in a periodic rise and fall of speed and a very unpleasant variation of light from the lamps to which the machine is supplying current. This hunting may be stopped by closing the stop-valve until the control of the engine is taken from the governor, and the speed of the engine is slightly below that for which the governor is set. When the load is steady the valve may be gradually reopened.

It is well always to keep a nearly constant pressure of steam up to the governor valve. In some cases a reducing valve is used, but, in the absence of this, careful regulation of the stop-valve will much assist the governor which will not closely control the speed under varying load if the steam pressure is also varying.

This regulation of the steam is much easier if a pressure gauge is fitted between the stop and governor valves.

Adjustment of governor.—A good way to adjust an engine governor is as follows. Overhaul the gear to see that the whole of the parts are well fitted and free, and that there are no flats worn on the ends of the short arms of the governor balls. Next try the governor valve. It should be free but sufficiently steamtight when closed to reduce the engine speed below the normal with full steam pressure and no load.

After reconnecting the parts, the engine should be started without load, and the speed increased by gradually opening the stop-valve until the governor balls fly out to their widest position. By means of a trammel and marks on a part of the engine frame and on one of the bell cranks of the governor gear, it may be seen when the governor balls reach this position. The corresponding speed should not be more than 5 per cent. in excess of the normal speed, full steam pressure being used. If necessary the tension of the governor spring

should be altered to adjust this speed. The rod connecting the governor valve to the gear should now be shortened by the adjusting nut provided, until the stop-valve can be fully opened without any further increase of speed. This adjustment insures that the permanent increase of speed when the load is gradually removed shall not exceed the limit allowed. The increase due to sudden removal of the load will depend on the relation which the moment of inertia of the fly-wheel and other moving parts of the engine bears to the time taken by the governor to close the valve.

When the load is put on, the speed will be reduced and the governor will open the valve until the steam opening is sufficient. The amount of the variation of speed for full load and no load will depend on the sensitiveness of the governor and on the steam pressure, a greater steam opening, and therefore a greater movement of the balls, being required when steam is low than when it is high. By small adjustments of the tension of the regulating spring the normal speed may be obtained for average load, and if the steam pressure is kept fairly constant the variation of speed due to change of load will be small.

Over-compounding.—It should be noted that to maintain a constant electrical pressure at the machine terminals, a perfectly compounded dynamo requires to be driven at constant speed. It is usual to slightly over-compound direct driven machines, so that the small loss of electrical pressure due to the fall in speed necessary to change the position of the governor when the load is increased, is compensated for by the increase of pressure produced by the increase of current flowing in the series coils of the machine. This action also tends to reduce the effect of the resistance of the leads between the dynamo and the lamps.

Effect of increase of temperature.—During the first few hours of running, the electric pressure at the dynamo terminals will fall gradually, due to the increase of resistance of the machine circuits as the temperature increases. After a time the temperature will become constant and no further variation of pressure will occur. It will generally be found that to maintain the full pressure the speed requires to be a few revolutions above the normal, the amount of increase depending on the temperature of the surroundings.

Electric motors.—The development of electrical science has now rendered possible the transmission in an efficient and economical manner of the power generated in the dynamo to electric motors in different parts of the ship, and this is taken advantage of to an increasing extent, and are of great advantage in confined spaces, where the heat of steam would be objectionable. It also reduces complication, as the wire leads can be run along in places where steam and exhaust pipes would be most difficult, and large holes in water-tight bulkheads for steam and exhaust pipes avoided. Electric motors avoid waste due to condensation, radiation and leakage in pipes, require little attention when running, are always ready for starting, and are extensively used on war-ships. New ships of the British Navy are being fitted with electric motors for ventilating and boiler-room fans, air-compressors, ice-making machines, coal whips, workshop engines, ammunition hoists, elevating and training guns, boat hoists and after capstans. They are still more extensively used in foreign navies, even steering gears there being often worked by electric motors, and in

many cases all the gun-working machinery, usually operated by hydraulic power, is also worked by electric motors. In such cases a very considerable addition is required to the electrical power, and four or more electric generating machines are fitted.

Oil-driven electric generating machines.—It is a great convenience when the ship is in harbour to be able to allow the fires to die completely out in all boilers, which allows of many necessary repairs to steam pipes and engines being effected. For this purpose oil-driven electric generating machines are being fitted for a proportion of the power in new British war-ships, and these engines are capable of using the same oil as that supplied for use in the boiler furnaces.

CHAPTER XXVIIIA.

INTERNAL COMBUSTION ENGINES.

Owing to the increasing use of internal combustion engines for various purposes in connection with ships and boats such as for electric light engines, boat propulsion, &c., and to its possibilities as regards the future, a brief description of such engines will now be given.

In the case of steam-driven machinery the fuel is burnt in a boiler furnace, and transmits to the water in the boiler a certain proportion of its heat of combustion. The water is converted into steam by the absorption of this heat, and, passing to the engine cylinders, does work at the expense of its heat and pressure energy, being finally exhausted into the condenser, and condensed into water by the extraction of heat by the circulating water. If it were at all practicable to pump the exhaust steam back into the boiler as steam, a great waste of energy would be avoided, since the heat given up to the circulating water by the condensation of the steam is wasted, and has to be supplied by the fuel in the boiler. It is, however, impracticable to deal with the exhaust steam in this way, and the loss (which is comparatively large owing to the large latent heat of steam) is inevitable.

Working fluids used.—If we were able to use a working fluid which did not change its state during the working cycle, then we should avoid the loss referred to above. Such a working fluid as air would satisfy this condition.

Now, the heating of a working fluid such as air in the same way as water is heated in a boiler, i.e. by heat transmitted through plates, would involve considerable difficulties; and if the engine were of large power, the bulk of the air-heating apparatus would be prohibitive. Such engines, called 'air engines,' have been made, but only of comparatively small power, and even these were very bulky. They have now been abandoned.

Air, however, in addition to being a fluid which does not change its state during the working cycle, is a supporter of combustion. If an inflammable gas or vapour be mixed with air in certain proportions, an explosive mixture is formed, the ignition of which causes an evolution of heat, and the products of combustion consequently rise in temperature and pressure; the effect is more pronounced if the mixture is initially compressed. If the mixture of combustible and air be formed within the working cylinder, and ignited, the products of combustion will rise in temperature and pressure, and will do work in expanding and driving the piston before them.

An engine working on such a principle is called an 'Internal Combustion Engine,' and two important properties of such engines are (1) the elimination of boilers and their equivalents, (2) the working fluid does not change its state, which increases the economy. Lately

great developments have been made in this type of motor, and no doubt further developments will increase their field of usefulness. It is not proposed in this work to go deeply into this class of motor, and only the general features will be described. The fuels used are generally either gases, vapours or liquids (generally oils obtained from petroleum), the engine being termed a gas, spirit, or oil engine according to the fuel used.

Cycle of operations.—Internal combustion engines use either the 'constant-volume' or the 'constant-pressure' cycle. With the 'constant-volume' cycle combustion of the fuel takes place at constant volume and is therefore more of the nature of an explosion; in the 'constant pressure' cycle combustion takes place gradually and at approximately constant pressure. The 'Otto,' or 'four-stroke cycle,' which is the one most commonly used with internal combustion engines, is a 'constant-volume' cycle. The Diesel engine, which is now being rapidly developed for marine purposes, works on the 'constant-pressure' cycle.

In the Otto cycle there is one ignition per cylinder every four strokes. On the first outward stroke of the piston the air and combustible are drawn into the working cylinder. At the end of the stroke the supply is shut off, and the contents of the cylinder are compressed during the return stroke into the clearance space, and are ignited at the dead centre. The products of combustion rise in temperature and in pressure, the volume remaining approximately constant during the combustion, and expanding, do work on the piston, this being the working stroke. On the return of the piston the products of combustion are discharged into the atmosphere through the exhaust pipe, and the cycle is completed. In some cases air only is admitted during the first, or suction, stroke, the combustible not being admitted until the end of the second or compression stroke.

A large number of engines work on the two-stroke constant volume cycle, which means that there is one ignition of the charge in every two strokes, and gas engines of fairly large powers are in successful operation on this cycle.

The Diesel four-stroke cycle differs from that of the 'Otto' cycle in having a very high compression, and the temperature due to the high compression is sufficient to ignite the fuel as it is injected at the dead centre. The products of combustion rise in temperature at approximately constant pressure so long as the fuel is injected, and expand. When the fuel supply is cut off, the products of combustion expand still further, work being done on the piston during the expansion. The other two strokes of the cycle are similar to those of the 'Otto' cycle. The Diesel engines which work on this cycle are interesting in that ignition devices are eliminated. Diesel engines have also been made to work on a 'two-stroke cycle.'

Theoretical diagrams of engines working on the 'Otto' and Diesel cycles are shown in Fig. 385. In the actual Diesel diagram the compression and expansion curves are not adiabatic. At the beginning of compression the air may be lower in temperature than the cylinder walls, and there may be a transfer of heat from the walls to the air. As compression advances the temperature of the air increases and heat is then imparted to the cylinder walls, and consequently the resulting

compression pressure will be less than that due to adiabatic compression. It is found, however, that during expansion the pressure is greater than that due to adiabatic expansion, although a considerable quantity of heat is being abstracted by the cylinder walls. This can only be due to the fact that combustion must be taking place practically throughout the expansion stroke.

Diesel engines are now being constructed to work on a two-stroke cycle, the operations, assuming the piston to be descending on its power stroke, being as follows :—Towards the end of its stroke the piston uncovers the exhaust ports on one side of the cylinder, Fig. 385A, and the burned gases escape through these ports to the exhaust pipe. The pressure in the cylinder is now approximately atmospheric, and further downward movement of the piston uncovers the air ports, which are opposite the exhaust ports in the Figure shown, through which a charge of compressed air or, as it is generally termed, 'scavenging' air, is admitted to the cylinder, which blows the remaining burned gases out through the exhaust. The piston on the return stroke first closes the scavenging air ports and then the exhaust ports. The cylinder now

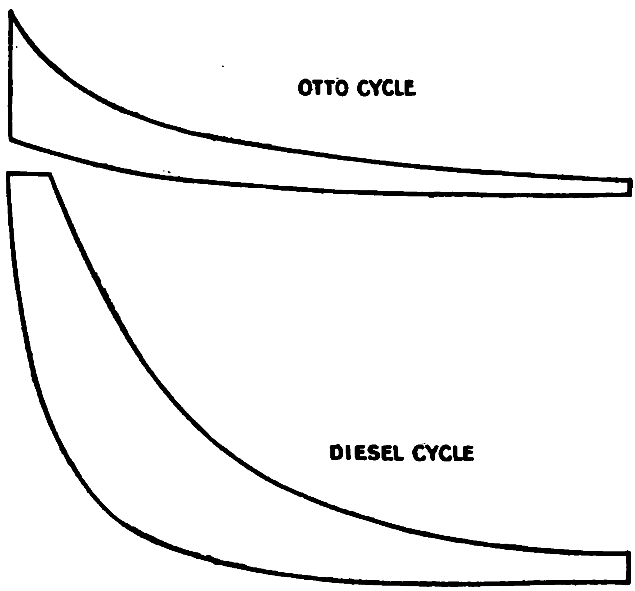


FIG. 385.

contains a charge of fresh air, and further upward movement of the piston compresses the air to about 500 lbs. pressure per square inch. As previously mentioned, this high compression raises the temperature of the air considerably, so that as the fuel spray is admitted, at the top of the stroke of the piston, combustion commences. The piston returns on its power stroke and combustion continues at approximately constant pressure until the fuel valve is closed. The products of combustion expand and do work on the piston and the cycle is repeated.

In some engines of this type instead of having air ports in the cylinder separate air scavenging valves, worked off the cam shaft, are fitted in the cylinder head.

Efficiency.—The higher the compression pressure in a cycle the greater the efficiency of the cycle. The practical advantages of high compression are (1) it increases the temperature of the charge before ignition and consequently ignition proceeds more rapidly; (2) less heat passes through the reduced wall surface at the highest temperature of explosion; (3) the clearance space being smaller the quantity of burned gases which remain in the cylinder is less, and it is possible, therefore, to obtain a purer explosive mixture; (4) the distance through which the flame must propagate is reduced.

As regards the cycles mentioned, viz., the constant-volume and constant-pressure cycles, assuming the same compression pressure, the former cycle is theoretically more efficient than the latter. The maximum pressure of the cycle is governed, however, by the pressure of compression, and, as the engine must be designed to withstand this maximum pressure, for practical reasons the pressure of compression is limited. From a practical point of view, therefore, in making a comparison of the two cycles it is better to assume the same maximum pressure. Making this assumption the constant-pressure cycle is then, theoretically, slightly more efficient than the constant volume cycle.

Apart from the question of design, the compression pressure is limited by the pre-ignition of the charge. Compression of the charge increases its temperature, and as all mixtures of fuel and air possess a critical temperature at which they will spontaneously ignite, this limit of compression is to be avoided. The only way to avoid pre-ignition in an engine using high compression is to compress the air separately and inject the fuel at the part of the stroke at which combustion is desired. It is here that the Diesel engine has an advantage over an engine working on the Otto cycle, as the former can use high compression pressures up to the practical limit.

In all internal combustion engines for marine purposes the cylinders and covers are water-jacketed, so as to keep the temperatures of these parts within reasonable limits. This is a source of loss, but is unavoidable. Another source of loss, and this is generally a large item, is the heat rejected with the exhaust gases.

In Chapter II. the losses which take place at each stage of the conversion of the heat energy into useful work have been analysed; it will be interesting to compare the losses in internal combustion engines:—

	Diesel engine. Per cent.	Gas engine. Per cent.
Work done	39	29
Heat lost in jacket water . . .	20	25
Heat lost in exhaust gases . .	35	40
Radiation, &c.	6	6

The figures given in the case of the Diesel engine are for an engine working on the four-stroke cycle. In the case of the two-stroke Diesel engine the work done should be approximately the same as that given for the four-stroke cycle, assuming that the scavenging of the exhaust gases is complete. As regards the other items, it is probable that the

heat lost to jackets would be slightly more than in the case of the four-stroke engine, and the heat lost in the exhaust gases slightly less.

Taking the mechanical efficiency for the four-cycle Diesel engine at 80 per cent., and for the gas engine at 85 per cent., the resulting efficiencies of propelling machinery for the two engines are :—

Diesel engine . . . $\frac{39}{100} \times \frac{80}{100} \times \frac{70}{100} = 21{\cdot}8$ per cent.

Gas engine . . . $\frac{29}{100} \times \frac{85}{100} \times \frac{70}{100} = 17{\cdot}2$ per cent.

It is seen that these efficiencies are very high compared with the reciprocating steam engine and boiler installation.

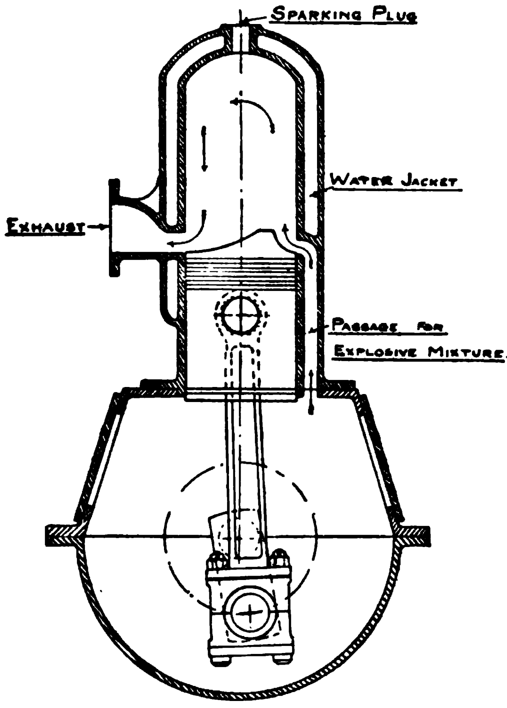


Fig. 385a.

The mechanical efficiency of a two-stroke Diesel would be less than a four-stroke engine on account of the extra work required to be expended for driving the scavenging air-pump. In the four-stroke engine, however, we have to overcome the friction of the several parts during the suction and exhaust strokes which do not occur with the two-cycle engine. It is probable that the mechanical efficiency of a two-cycle engine would be between 70 and 75 per cent., but even so the resulting efficiency is high compared with the best type of marine steam engine.

Gas producers, &c.—Gas engines are sometimes designed to use ordinary coal gas, but by far the largest number are operated by what is called 'producer gas.' Coal gas is a mixture of hydrogen about 50 per cent., marsh gas about 33 per cent., carbon monoxide about 13 per cent., illuminants and nitrogen about 4 per cent., and has a calorific value varying from 500 to 700 British thermal units per cubic foot. Producer gas is made by causing a mixture of air and steam to be drawn, or forced under pressure, through a deep layer of incandescent coal or coke contained in a 'producer'; the steam is decomposed, forming hydrogen and oxygen, the latter uniting with carbon in the lower part of the fire forming CO_2, which, passing through the incandescent fuel above, takes up more carbon, forming CO. The oxygen of the air is similarly combined, and the gas collected in the upper part of the producer consists of hydrogen 10 per cent. to 25 per cent., carbon monoxide 10 per cent. to 25 per cent., nitrogen 50 per cent., carbon dioxide 5 per cent. to 20 per cent. The calorific value is about 140 British thermal units per cubic foot.

The gas is cooled on leaving the producer by passing through or around tubes surrounded by or containing water, from which the necessary steam is generally obtained, and is cleansed by passing through scrubbers containing coke, shavings, or sawdust. The gas cleaning apparatus is very bulky when the coal used is of a bituminous nature, but not unduly so when anthracite or coke is used in the producer. Dust and tar are the two great obstacles to the successful working of gas engines, and it is to eliminate these products that an adequate cleaning plant has to be provided.

When the air and steam are forced under pressure through the producer, the plant is said to be of the 'pressure' type, and a separate small boiler is usually needed to generate the necessary steam and to compress the air; when the air and steam are drawn through the producer it is said to be of the 'suction type,' and the separate boiler is usually dispensed with.

On the two types the suction plant is considered preferable for the following reasons:—(1) The use of a separate steam boiler is obviated. (2) The gas is at less pressure than the atmosphere, and consequently any leaky joints will cause air to leak into the system instead of gas leaking out.

The second point is of considerable importance in marine work, where the producers would most probably be installed in confined spaces. Under these conditions the leakage of a gas containing a high percentage of poisonous carbon monoxide would be dangerous.

Other fuels used.—When a spirit is used which is very volatile and of a homogeneous nature, it is generally capable of being easily applied to internal combustion engines. A current of air passing on its way to the cylinder will saturate itself with the vapour of the spirit if it be led over a finely divided jet of the liquid. This principle is made use of in petrol-driven motor-cars, launches, &c., and is the simplest method of forming an explosive working mixture, since no elaborate apparatus is needed to vaporise the liquid. Spirits which are so volatile are, however, dangerous for use on board ships, and in order to safely stow and use large quantities of such spirits afloat very

elaborate precautions would have to be taken, and the general use of such fuels afloat could not be recommended.

Homogeneous liquids, such as lamp oils, etc., which are not sufficiently volatile to be converted into explosive mixtures when air is passed through a finely divided jet of the liquid, are much safer than spirits, and generally require some application of heat to convert them into vapours; in this state the fuel can be carried by a current of air into the cylinder, forming there the explosive working fluid.

Vaporisers.—In most cases the fuel is delivered into a vaporiser external to the cylinder, which is heated by a lamp at starting, and either by a lamp or by a jacket of hot exhaust gases when the engine is running. The heat converts the liquid into a vapour, and either the whole or a part of the air supply is drawn through the vaporiser, sweeping with it the vapour into the cylinder. In certain cases the vaporiser is designed to form part of the combustion space, being always in communication with the cylinder. It is heated at starting by means of a lamp, but when the engine is working it is maintained at a temperature sufficient to vaporise the oil by the heat of the previous explosion. The liquid is usually injected direct into the vaporiser in this type, which is advantageous in that the liquid is untreated by heat, and is therefore quite safe, until it is inside the combustion space. Such a type of vaporiser enables the engine to deal with oils of very high flash point and of complex chemical structure, and in engines fitted with such vaporisers it is not necessary to have a refined, i.e. a fairly homogeneous oil. The Hornsby-Akroyd engine works on this principle.

Ignition, in the ordinary type of internal combustion engine, is generally effected by either an electric spark or by a heated tube of special metal or porcelain, whilst in some cases the ignition is automatic, taking place at the desired moment by the contact of a properly proportioned explosive mixture with a red-hot surface, maintained in this condition by the heat of explosion.

Scavenging.—As already mentioned, engines are constructed working on the two-stroke constant volume or constant pressure cycles, and in these cases special means have to be provided for supplying the scavenging air. The method usually adopted is one of the following:—

(1) Using the crank case as a compressor.

(2) Enlarging the impulse piston at its lower end and arranging the air cylinder concentric with, and below, the impulse cylinder.

(3) Separate scavenging pumps.

The scavenging medium used is either the fuel mixture, that is the explosive mixture of fuel and air, or air only.

In small engines working on the constant volume cycle method (1) is generally used, with the fuel mixture as the scavenging medium. This method, apart from the fact that a certain amount of fuel is lost during the scavenging process, is not so efficient as the second or third methods. Fig. 385A shows a diagrammatic sketch through the cylinder and crank base of such an engine. During the up-stroke of the piston the fuel mixture is drawn into the crank case through a non-return valve, not shown in the figure. As the piston descends the charge within the crank case is compressed, and when the piston uncovers the scavenging port this compressed charge enters the cylinder. In order

to make the scavenging as complete as possible, a lip is cast on the piston, which directs the incoming mixture to the top of the cylinder, as shown by the arrows.

The second method, although more efficient than the first, makes the engine comparatively higher, and the parts are not so accessible for examination and adjustment.

The third method of separate pumps, with air as the medium, is the one generally employed with Diesel engines for marine purposes. By this method an excess of scavenging air can be obtained which is needed to ensure the scavenging of the burned gases being as complete as possible.

It is essential for the successful working of the two-cycle engine that the working cylinder should be thoroughly scavenged of burned gases. If this is not carried out the power of the cylinder will be adversely affected. Especially is this important in the case of the engine using the fuel mixture as the scavenging medium, for if a comparatively large proportion of burned gases are allowed to remain in the cylinder the volume of the fuel mixture per stroke will be decreased, and there may be some difficulty in igniting it efficiently, both of which will affect the power. In extreme cases, if a comparatively large quantity of the burned gases are allowed to remain in the cylinder the fuel mixture may fail to fire at the desired moment, and there is also the possible danger of pre-ignition.

The scavenging pressure required depends upon the size of the air ports or valves. The pressure should be as constant as possible during the process of scavenging, and it will be seen that this is not obtained in the case of method (1). For complete scavenging, the burned gases should be ejected from the cylinder in one mass and not broken up by the incoming scavenging medium, and it is for this reason that the pressure of the latter is kept as low as possible.

In the Diesel engine the scavenging air pressure used varies from 3 to 7 lbs. per square inch. Some makers prefer to fit one double-acting scavenging pump for each set of engines, whilst others fit two or more for engines of large size. The minimum volume of the scavenging air pumps necessary for efficient scavenging is not yet definitely settled, but in some designs this volume is made 1·8 times the volume of the working or impulse cylinders, and in others it is as low as 1·25. It is essential that these pumps should be no larger than is actually required on account of the work lost in driving them.

Diesel engine.—The principle of this engine has been described above under the heading of 'Cycle of operations.' The compression pressure has to be about 500 lbs. per square inch to obtain the necessary temperature to burn the fuel. The high pressure used necessitates the working parts being designed of sufficient strength to withstand it, and consequently this type of engine is rather heavy, although Diesel engines of very light design are being constructed on the Continent. The fuel is blown in by compressed air at about 900 lbs. per square inch; thus an air compressor forms a necessary and important part of the engine.

A section through a cylinder of a high-speed single-acting four-cycle Dieselengine is shown in Figs. 385B and 385C. The fuel, exhaust and air valves are actuated through bell-crank levers by cams on the cam shaft.

The air inlet and exhaust valves and their seatings are generally arranged so that they can be readily removed for cleaning and examination. The cam shaft is driven from the main engine crank shaft through gearing—the ratio of speed of crank shaft to cam shaft being two to one.

The piston is made of special cast iron to withstand the high temperatures obtained, and is fitted with Ramsbottom rings as shown. It is very necessary that these rings should be tight to ensure getting the necessary compression pressure. The cylinders and covers are water-jacketed, the cooling water being circulated by means of a pump driven off the engine shaft.

Separate fuel pumps, worked off the cam shaft, are usually fitted for each cylinder. The fuel is delivered to the fuel valve casing, and is then forced through the pulveriser by means of high pressure air, entering the cylinder in the form of a finely divided spray. The high pressure air is obtained by means of a two- or three-stage compressor driven from the engine shaft, cooling arrangements being provided between each stage. The air from the compressor is discharged into a blast receiver, blast pipes being then led from this receiver to the various fuel valve casings of the impulse cylinders. The high pressure compressors are usually designed to charge the starting reservoirs also.

In addition to the air inlet, exhaust and fuel valves in the cylinder head, some cylinders, if not all, of a set of engines are fitted with an additional valve for admitting compressed air for starting purposes. These valves are worked off the cam shaft by cams and levers, and can be brought into action when required. The fuel supply to cylinders, which are fitted with starting valves, is usually shut off when starting air is being used. As soon as the remaining cylinders pick up the firing the starting air is shut off, the starting cylinders and the fuel supply to these cylinders brought into action.

The principal bearings are fitted with forced lubrication. In small engines the piston usually obtains sufficient lubrication from the splash of the oil emerging from the crosshead pin, but in large engines separate adjustable sight-feed lubrication is sometimes fitted to this part.

The construction of the remaining portions of the engine, other than those shown in Fig. 385B, is generally similar to the ordinary steam practice.

Two-cycle, single-acting Diesel engine.—Fig. 385D shows a section through a two-cycle Diesel engine. The several operations of the two-cycle engine have already been described. With this engine we obtain an impulse every revolution, and therefore the power developed is about double the power developed in a four-cycle engine of the same dimensions. In the Figure shown, apart from the starting valves, which would be similar to those already described for the four-cycle engines, it will be seen that the only valve in the cylinder head is the fuel valve, the exhaust and scavenging ports being cast in the cylinder. Some engines, however, of large powers have two or more scavenging valves in the cylinder head instead of scavenging ports.

The piston is of trunk form and the gudgeon pin is secured inside it as shown. This method is adopted for small powers, but for higher powers a piston-rod and crosshead are usually fitted. It is found necessary, beyond a certain power per cylinder, to resort to cooling the pistons of both two-cycle and four-cycle engines. With small engines,

however, of the single-acting type this is unnecessary, as it is found they are kept sufficiently cool by contact with the water-cooled cylinder walls, and also being open to the atmosphere on one side.

The advantages of the two-cycle engine compared with the four-cycle engine of the same revolutions and power are :—

(1) The power can be obtained with a smaller number of cylinders, the uniformity of twisting moment remaining approximately the same.

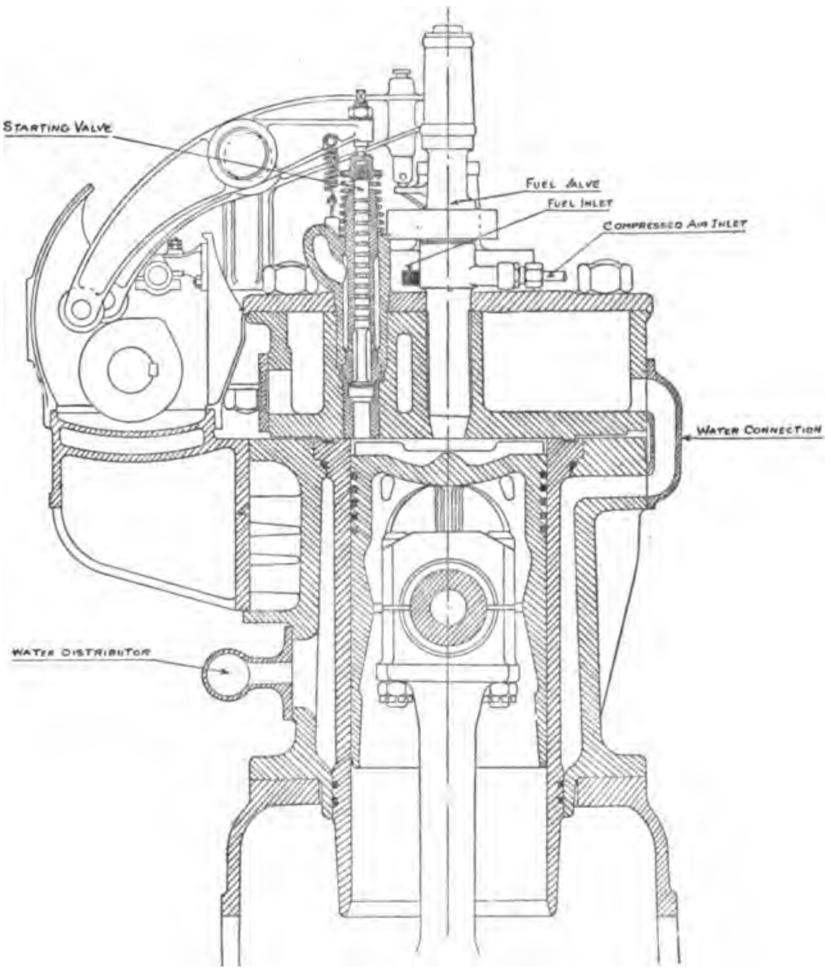


FIG. 385B.

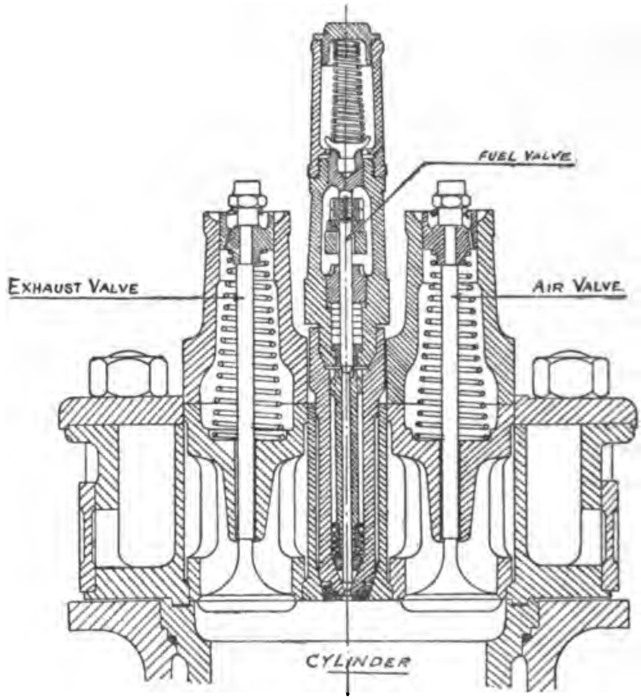
(2) The overall length of the engine will be less.

(3) The weight of the engine will be less, although scavenging air cylinders are necessary.

(4) The valve gearing and reversing arrangements can be made more simple.

The disadvantages are the additional scavenging pumps required, which slightly reduce the mechanical efficiency of the engine, and, the period of the heat cycles being half that of the four-cycle engine, the difficulty of cooling the parts efficiently, and consequently the possibility of piston troubles, will be increased.

Two-cycle, double-acting Diesel engine.—Double-acting engines of the Diesel type of comparatively large powers are being constructed. With this type of engine scavenging, fuel and starting valves are fitted



SECTION THROUGH CYLINDER COVER
SHOWING VALVES

FIG. 885c.

in the top and bottom of the cylinder. The exhaust ports, uncovered by the piston, are in the middle of the length of the cylinder. The piston is of box form and a piston rod and crosshead are fitted. It is necessary in this case to cool the piston and piston rod, and a stuffing box is of course necessary where the piston rod passes through the bottom of the cylinder. This stuffing box is sometimes connected by pipes to the suction side of the scavenging pumps in order to prevent the possibility of gases leaking through the box into the engine room.

Water is the best cooling medium for the piston and rod, but oil is very often used so as to avoid the danger of water becoming mixed, through leaks in pipes and connections, with the lubricating oil of the engine and causing trouble with the bearings.

The two-cycle, double-acting engine gives two impulses during each revolution like the ordinary steam-engine. The power can be obtained with a smaller number of cylinders than with the single-acting type, but the engine will be higher for the same stroke and cylinder diameter. The additional valves in the bottom of the cylinder and the fittings for operating them, the stuffing boxes and the cooling of pistons and piston rods, which may not be necessary with the single-acting type, add to complications, and therefore the question as to whether

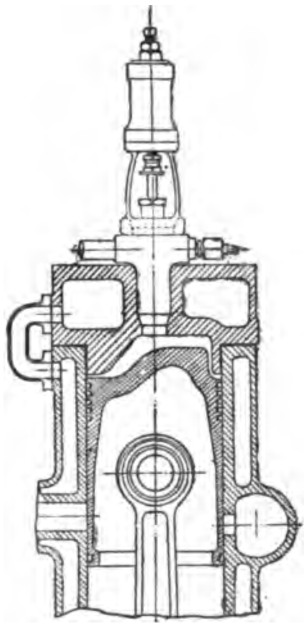


FIG. 385D.

single- or double-acting engines should be fitted for any purpose requires careful consideration.

Internal combustion engines for propulsion.—The special conditions to be satisfied by a successful marine oil or gas engine are much the same as those possessed by the present steam engines, viz. :—It must be capable of reversing the motion of the propeller, and being stopped or started quickly either ahead or astern. It must be capable of being promptly set and kept at any desired number of revolutions between dead slow and full speed. All working parts must be readily accessible for overhauling and capable of being promptly and easily adjusted. The engine must be economical in fuel, especially at its ordinary working speed.

The engines should be balanced, and the noise from the working should be such that for boats carried by warships it would not preclude the use of these boats for night attacks. The most important difficulty is to obtain a slow speed. Considering one quarter of full speed as dead slow, this would mean one sixty-fourth of full power, or the mean effective pressure would have to be about one-sixteenth of the mean effective pressure when working at full power. Thus this slow speed, which is very necessary for boats which have to manœuvre in crowded anchorages, is not usually obtained.

Oil engines of comparatively small powers have been installed in numerous small craft for various purposes, the astern movement being obtained by means of a reversing propeller or with a reverse gear and solid propeller. For large powers, however, this method of obtaining astern movement is not satisfactory and it becomes necessary to make the engine itself reversible.

No very large gas engines have yet been applied to marine propulsion, but considerable progress has been, and is being, made in the application of the reversible Diesel engine for this purpose.

Reversible four-cycle Diesel engines have been fitted in ships and are running satisfactorily; the two-cycle reversible engine, however, is the type generally adopted for marine work, the reversing arrangements with this type being less complicated. The number of cylinders should be such that the necessity for fitting a fly-wheel should be avoided, and with a four-cycle single-acting engine it would appear that this number should be not less than six, and in the two-cycle single-acting engine not less than three. The larger the number of cylinders the smoother the engines will work, but at the same time the space occupied will be greater, also the number of working parts, and consequently the work of examination and refitting required in connection with them, will be increased.

In considering the number of cylinders to be employed with Diesel engines, the maximum power to be developed in one cylinder should not be overlooked. There is no actual experience at present with the running of Diesel engines developing in one cylinder as much as 1,000 H.P., although it is understood that engines developing 300 to 500 H.P. per cylinder are running and have given no trouble, and also that an experimental single-acting two-cycle cylinder of 1,200 H.P. has been constructed on the Continent. As regards the two-cycle double-acting engine an experimental cylinder of 2,000 H.P. is being constructed.

In all cases high pressure air is used for manœuvring purposes, the air being supplied by two- or three-stage compressors driven by the main engines, which also supplies the blast air for injecting the fuel. The starting air is stored in a number of steel air bottles, and the quantity of air stored depends upon the number of reversals it is desired to arrange for. For large engines, in addition to the air-compressor driven from the main engine, a separately driven auxiliary high pressure air-compressor is fitted for augmenting the supply of compressed air when manœuvring.

In some designs all impulse cylinders are fitted with starting air valves, but in another design, although comparatively small powers have as yet been constructed, the starting air is admitted to the two

scavenging air-pumps, the cranks of these pumps being at right angles. The portion of stroke during which the starting-valves are open depends upon the number of cylinders on which starting air can be used.

In starting, the valves are put into action by means of a single wheel or lever. The first part of the movement releases any pressure of air or gas in the cylinders which would tend to prevent easy starting. Further movement admits air to all cylinders which are in a position to receive it—depending on the direction it is desired to move, i.e. ahead or astern. A still further movement cuts off the air supply from certain cylinders, and fuel is admitted to them at the correct moment. When these cylinders pick up the firing the wheel is moved again and the starting air is cut off the remaining cylinders, and all cylinders are then in action.

The cam shaft is driven from the main engine shafting by suitable gearing, and to bring the cams into the correct position for moving ahead or astern means are provided for giving the cam shaft a slight angular moment relative to the main engine shafting. Separate ahead and astern cams for all valves may, however, be provided, in which case a lever is fitted to bring these into action as desired.

The regulation of speed in marine engines is important, and is usually arranged for by operating on the suction valves of the fuel pumps so as to limit the quantity of oil discharged into the cylinders per stroke. A further reduction in speed could be obtained by slightly reducing the blast pressure. It is stated that the slowest revolutions possible with certain types and sizes of Diesel engines is about one-fifth the revolutions at full power, although one-fourth appears to be the generally accepted figure.

When a ship is fitted with oil-driven propelling machinery special arrangements have to be made for working the auxiliary engines. Some engines, such as the electric generators, could be oil driven, but it would be necessary for other auxiliaries to be driven either electrically or by compressed air, unless oil-fired auxiliary boilers were arranged for.

Steam would be necessary for distilling purposes, although it is probable that the exhaust gases from the main engines could be utilised in an evaporator adapted for the purpose. The latter, however, would only be available at sea, and in large vessels, especially warships, steam for the evaporating plant would be necessary, in which case it would probably be an advantage to drive most, if not all, of the auxiliaries by steam.

It has been suggested that the heat of the exhaust gases from the main engines of an oil-driven vessel might be utilised in raising steam in an auxiliary boiler, but it is very doubtful whether the steam generated would be sufficient for the purpose intended. If this method proved efficient at sea additional means would have to be provided for supplying the auxiliary power for use in harbour.

General remarks.—The great advantage of the internal combustion engine is its economy as compared with the steam-engine. Small oil engines using ·8 of a pound of oil per B.H.P. per hour are of ordinary occurrence, whilst in the Diesel engine the consumption for comparatively large powers is about ·45 of a pound of oil per B.H.P.

Gas engines can be obtained using in the producer less than 1 pound of coal per B.H.P. per hour, whilst ordinary marine steam-engines of large powers rarely use less than $1\frac{1}{2}$ pound of coal per I.H.P. per hour.

The economy of the oil-engine compared with the steam-engine may be very marked when used for intermittent work or such as would render the banking of fires necessary in a boiler.

The further advantages of a Diesel-driven ship over the steam-driven vessel of corresponding power are :—

(1) For the same weight of fuel carried the radius of action would be increased.

(2) The space required for the machinery would be less, although it is probable that the height of the Diesel engines, depending on the type fitted, would be slightly greater than reciprocating steam-engines.

(3) There would be a slight saving in the weight of machinery. This saving would be increased if the anticipated weights of certain designs of Continental Diesels are realised.

(4) Fuelling the ship would be simplified considerably and would be a much cleaner and more rapid operation. The fuel could be stowed in spaces which would otherwise probably not be utilised.

(5) Reduction in the stokehold staff, although the number of mechanics would have to be increased.

(6) If under-water exhaust proved possible, with large powers the absence of funnels would be an advantage in warships.

The disadvantages, however, are :—

(1) The high price of fuel oil and the limited production compared with coal.

(2) Difficulty in obtaining supplies at all ports at which a ship may call.

(3) Greater initial cost of the oil-engine installation.

(4) Vibration of the machinery compared with turbine vessels.

(5) The lowest possible speed is greater than with the steam-engine.

(6) Compared with the steam turbine, the reversion to reciprocating engines, together with the larger number of small working parts, the delicacy of the valve gearing adjustments and possible piston troubles, &c., involves much greater attention on the part of the engine-room staff to keep in good order.

(7) The satisfactory working of the main engines is dependent on such auxiliaries as fuel pumps, high pressure compressor, scavenging pumps, &c.

(8) The main engines are dependent on compressed air for starting and reversing.

Should the Diesel engine prove thoroughly reliable in practice, that is, as reliable as the steam-engine has become, after many years of improvements in design and workmanship, the disadvantages, with the exception of (2), especially in the case of warships, could probably be neglected in view of the advantages obtained with this method of propulsion.

As regards (2), the Diesel engine is capable of using almost any fuel oil, and is in a better position than an internal combustion engine which has to use a special fuel or refined oil. In spite of this,

however, the fuel oil supply is limited and not obtainable at every port, although the number of ports at which fuel oil can be obtained is increasing, and therefore this is a serious drawback to the use of the Diesel, or any other type of oil-engine, for the propulsion of vessels which are expected to go to any part of the globe.

A few oil engines capable of using residual fuel oils of about 200° F. flash point are fitted in H.M. ships, for driving dynamos, in order to obtain experience with this type of motor. The provision of such engines, if reliable and of sufficient capacity, enables steam to be put down in harbour without interfering with the electric lighting and ventilation of the ship or the performance of any other electrically operated machinery.

CHAPTER XXIX.

RAISING STEAM AND GETTING UNDER WAY.

In this chapter the procedure of raising steam in the boilers and getting the engines under way will be described. It will be assumed that the engines have not been worked for some time, the boilers being empty.

Filling the boilers.—When preparing to raise steam, the boilers are filled with water to some distance above the ordinary working level, say to about three-fourths the height of the gauge glass, to allow for the losses that occur during the warming of the engines. In modern vessels clean fresh water should always be used unless absolutely unobtainable. Salt water can be used in ships with jet condensers, and is also permissible in some of the older vessels in which the steam pressure is less than 90 lbs. per square inch, and in which salt water make-up for boiler feed is a necessity. When clean fresh water is not available, every care should be taken to avoid the necessity of filling the boilers from shallow depths, or for using impure water of any kind, and if in harbour the boilers should be filled at high water, but in no case should the water be taken from refitting basins.

Boilers may be filled with fresh water, (*a*) by means of a hose from the jetty, or water-boat alongside, through one of the upper manholes, (*b*) by connecting the hose to the boiler-emptying valve, or (*c*) by the auxiliary feed engine from the reserve fresh-water tanks.

When ships are without steam power, either (*a*) or (*b*) is adopted, and when steam power is available in the ship (*c*) is the most usual plan, the supply to the auxiliary feed-pumps being maintained in the reserve fresh-water tanks by the ship's injectors. In the first case (*a*) is the more expeditious method, as it allows a full ingress for the water, at the same time providing full egress for the air, and it is the method generally adopted.

In plans (*b*) and (*c*) provision must be made for the escape of air by opening the safety valves, air, water-gauge drain, test, and hydrometer cocks, the latter three being closed as the water reaches their respective levels, the others when the water has reached the required level.

Boilers may be filled with salt water by means of the surface blow-out apparatus if the water level of the boilers is below the sea level, as is usually the case; or if this be not sufficient, it may be supplemented by means of a hose through one of the upper manholes, or connected to the boiler-emptying valve, the supply being procured by

the ship's hand-pumps through the fire main; or when steam is available, by the auxiliary feed engine from the sea direct.

Raising steam.—Prior to lighting fires the following points should be attended to:—

Boiler room procedure.—Fill the boilers as described above, then having removed or opened the funnel covers, the funnel guys should be slackened, to allow for the expansion of the funnel; the cowls on the downtake tubes should be worked and trimmed to the wind; and any dampers in the uptakes should be opened wide. Any forced draught fittings, including the air-pressure gauges, should be examined, and there should be no combustible material in proximity to the boilers. A reserve of fresh water should, if practicable, have previously been provided in the reserve tanks. The main and auxiliary stop-valves on the boilers should be opened sufficiently wide to allow a free passage of steam to the engines, the necessary amount being determined by experience, also any valves in the steam pipes leading to the auxiliary engines to be used should be opened wide, except the valves at the engines themselves. The safety valves should be closed, so that the heated air and vapour will pass from the boilers to the main and auxiliary steam pipes, gradually raising their temperature.

Engine room procedure.—The following valves in the main steam pipe should be opened, viz. bulkhead stop-valves; intermediate screw-down valves; regulating and manœuvring valves; connecting valve in pipe between engine rooms; valves to cylinder jackets, main circulating engine, starting engine, starting valves, and steering engine, if steam for the latter is to be taken from main steam pipe.

The silent blow-off valve should be closed. The jacket drain valves should be opened to the condensers, and the cylinder and slide-casing drains to the bilge.

The exhaust valves on the auxiliary condensers and the inlet and discharge valves of the main and auxiliary circulating pumps should be opened, the air-cocks being kept open till each condenser is full of sea-water. As soon as any fresh water accumulates in the condenser, test its freshness, to ascertain whether the tubes and packings are free from leakage.

All crank-pits and working parts generally should be examined for obstructions, and the main engines turned through a complete revolution by the hand-turning gear, after which the turning gear should be disconnected and secured. It is also advisable to run the links from full gear ahead to full gear astern by hand before connecting the steam starting engine.

All feed-tank fittings should be examined, and the suction valves for the main and auxiliary feed engines opened. The indicator and revolution counter gear should be connected and adjusted. Any relief cocks for backs of slides and balance pistons should be opened.

Generally.—It should have been ascertained that all joints are made and glands packed; all securities for working parts and holding-down bolts are in position; all lubricators are connected and clear, the oil lubricators being filled and the necessary worsteds ready; all pressure and vacuum gauges connected, with their shut-off cocks and cocks at gauges open; all drain-cocks on the various steam and exhaust valve boxes and pipes are open and connected to the drain tank direct, the

direction cocks on the steam traps being so arranged. All auxiliary engines necessary for getting under way and steaming should be turned through one revolution by hand; all steam and exhaust connections, except those on the engines themselves, should be open; and all oil lamps, including those for pressure and water gauges on the boilers, should be properly trimmed for use before the dynamo engine can be started.

Laying fires.—The method of proceeding will depend on whether steam is urgently required or otherwise, but to be definite we will describe a normal case of raising steam in a three-furnace tank boiler in which, to insure a gradual increase in the temperature of the various parts, and to diminish, as far as practicable, the stresses due to their expansion, the time allowed is rarely less than eight hours. The three furnaces would often be lighted together, and no fixed rule can be given as to the absolutely best procedure in this respect, but the practice of first lighting one furnace only, and after an interval dealing with the others, is perhaps the best plan, and this will be described.

A measured quantity of coal is placed on the floor-plates in front of each of the boilers which are to be used; the bars of each furnace should then be '*primed*'—i.e. covered throughout with a layer of average-sized pieces of coal. One furnace, usually the lowest, is next '*wooded*'—i.e. pieces of firewood, so arranged as to facilitate the access of air to all parts, are placed at the furnace mouth, over a bed of oily waste, shavings, &c. The wood is next '*topped*'—i.e. the space between the wood and the furnace-crown is practically filled with pieces of hand-picked coal.

Lighting fires.—To light the fires the oily waste, &c., is kindled, at the same time the furnace door is left wide open, and the ash-pit doors closed, so that a good draught is insured through the fire, and the flame is carried over the coal laid on the furnace bars and tends to ignite it. Both the furnace and ash-pit doors of the other furnaces are kept closed to prevent the access of cold air. Lighting the fire in one furnace tends to set up circulation in the water, and promotes uniformity of temperature throughout the mass.

As the fire burns it is continually topped with hand-picked coal, and after about two hours there will be a fairly substantial fire at the mouth of the furnace. After this has been done, it is usual to light one or both of the wing fires from the fire in the central furnace. These fires are made at the front of the bars and constantly topped with hand-picked coal, the furnace and the ash-pit doors of these furnaces being arranged similarly to those of the middle furnace. If only one wing furnace is lighted at first, the other would be similarly treated about one hour afterwards.

Spreading fires.—After about four hours the centre fire is *spread*—i.e. the fire which till now has been on the front of the furnace is spread over the partially ignited coal on the remainder of the bars—after which this furnace door is closed and ash-pit door opened, so as to admit the air underneath the furnace bars and so promote the combustion of coal throughout. The other fires are similarly treated at about the fifth or sixth hours, at the discretion of the person in charge.

About this time the water should be boiling and pressure begin to show in the gauges. The rate of combustion can now be regulated by

the amount of opening given to the draught-plates, and depends on the time at which steam is required to start the engines.

If steam be required for an emergency all the fires would be lighted simultaneously and spread earlier than described, but rapid raising of steam in water-tank boilers is to be deprecated, except in cases of absolute necessity.

When steam shows in the boilers. Boiler room procedure.—Soon after steam pressure begins to show on the pressure gauges, the safety valves should be lifted from their seats for a short period, and this should be repeated when the steam pressure is near its maximum. Each water-gauge cock on boilers and separators should be worked and test cocks tried, to see they are clear, and the water gauges tested frequently whilst the pressure is rising. The pressure gauges should be watched to see that each shows practically the same pressure, which should be the case if all the main stop-valves and the pressure gauge cocks are open and the gauges in good order.

The openings allowed for the main and auxiliary stop-valves depend on the amount of steam required and on experience, but they are sometimes opened fully, if there is no tendency in the boilers to prime. Frequently also, in the case of long stokeholds, it is found beneficial to open those nearer the engines less than those more distant, so as to equalise the supply of steam from each boiler, and to enable this to be properly effected the initial opening of the valves should not be great.

In the engine room.—The regulating valves should be closed and the manœuvring valves regulated so that the cylinders may be warmed up gradually. The main and auxiliary circulating pumps should be started as soon as possible, to prevent the condensers becoming heated from the drainage and exhaust steam; also the auxiliary air-pump, to keep both the auxiliary and main condensers free from condensed water. This latter duty is in some cases performed by some other engine.

As soon as the pressure is sufficient the drain-tank engine should be started, and the direction cocks on the steam traps of the steam and exhaust pipes should be set to allow the drainage to pass through the traps. Special care should be taken with the drainage of the main and auxiliary steam and exhaust pipes, and the separators, where so fitted, should also be kept well drained. Accumulated water in pipes is a common cause of accident. It may break down the engine by entering the cylinders, or, by being set in motion by the steam, cause fracture of a valve box or pipe, should its motion be suddenly arrested or changed. This water hammering action must be carefully guarded against.

The main and auxiliary feed-pumps should be tested by pumping from the feed-tanks, and the latter also from the reserve fresh-water tanks, and delivering into the boilers. They should be again tested against the full steam pressure before the main engines are reported ready.

Any other necessary engines in the stokehold should be worked for trial, and it may be necessary to keep the stokehold fans slowly revolving for ventilation, but they should not, except in cases of necessity, be used for rapidly urging the fires and quickly raising steam.

Warming the engines.—The cylinders should be warmed as

allowing the links to pass frequently from full gear ahead to full gear astern, at the same time permitting a very small amount of steam to pass through the manœuvring valves, and thus through the cylinders to the condenser; (c) by occasionally opening the auxiliary starting valves. The cylinder and slide casing drains should be used to keep the engines clear of water, but unnecessary waste of steam should be avoided.

As regards (a), the jacket steam and collector drain valves should be regulated to keep the proper pressure in the jackets, whilst at the same time insuring their efficient drainage; e.g. the valves should be so set that the collector is kept about one-third full of water by the gauge glass. (b) This is easily done in the case in which an 'all-round' starting gear and engine is fitted, as the starting engine can be set slowly rotating continuously in the same direction, care being taken that the quantity of steam admitted at any time by the manœuvring or auxiliary starting valves is not sufficient to move the engines. If the engine be fitted with independent linking up gear on the various cylinders the gear should be set in the full cut-off positions. (c) When the starting valves are fitted to admit steam to the receiver pipes they should be kept slightly open. When fitted direct to the cylinders, their direction will require reversal to top and bottom of cylinder alternately.

The electric-light machinery should be started on the circuits in the various machinery spaces as soon as there is sufficient steam. The glands on the stern tubes should be slackened to permit of a slight leakage through them, the fire and bilge engines being used on the bilges as necessary, and the crank-pit save-all pump started if separate.

The main water-service valves should be opened, and just before starting, water should be allowed to circulate through the backs of the ahead crosshead guides.

The telegraphs and voice pipes should be tried as to working and correctness, and the steam connections to the siren and whistle should be opened, the pipes drained, and the whistle and siren tried and left with their steam connections open. Funnel guys should now be adjusted.

As soon as there is sufficient steam pressure the steering engine should be put in gear. Having first ascertained that the locking gear is out and the rudder otherwise free, it should be first worked at the engine itself by putting the rudder to both extreme positions, leaving it at the middle position. The controlling gear for the deck steering position should then be connected, and the engine run hard over each way from the bridge. Both rudder and steering wheel should be placed in the amidship position before connecting up.

The engineer in charge, having satisfied himself that the above particulars have been attended to, and that steam is raised to about the required pressure, with the fires properly burnt through, the correct water level in the boilers, and the feeding arrangements satisfactory, and also that the cylinders are thoroughly warmed and drained, should, about ten minutes before the main engines are required, see that all worsteds and other main engine lubricating arrangements are set in

engines, to insure that they are under proper control, both in the ahead and astern directions.

Starting and stopping.—Although from deck considerations it is often of importance to move the engines as little as possible before unmooring, it is nevertheless most desirable that the engines should make at least one complete revolution in each direction under steam before they are reported ready for getting under way, and, if there are no special reasons to the contrary, they should be allowed to make a few complete revolutions in each direction in order to ascertain that they are under control and can be readily handled before weighing anchor or slipping moorings. This trial should not be sufficient to produce any way on the ship.

Having received permission and seen that everyone is clear of the engines, drain the main cylinders and slide-casings and the cylinders of the starting engine, and work this engine for putting the links into full gear ahead. Open the manœuvring valve, but not too wide, when, provided the high-pressure crank is not on the dead centre, if the high-pressure slide-valve is in such a position as to admit steam to that cylinder, and the cylinders are properly warmed and drained, the engines should start. If the high-pressure slide-valve is closed, or practically so, the auxiliary starting valves must be used, when no difficulty should be experienced in starting. The manœuvring valve should now be closed, the links be put into the full astern position, the manœuvring valve opened as before, and the engines allowed to revolve in that direction for a few revolutions, when the manœuvring valve should be closed and the links centred, so as to stop the engines.

After stopping it should be noted to what degree the main condenser retains the vacuum shown on the gauge, which will give an indication of its freedom or otherwise from air leaks. The revolution tell-tale should now be in gear. The engines should now be reported ready for getting under way, and the cylinders and slide-casings then kept drained as necessary, the proper pressures being maintained in the cylinder jackets and their drains suitably adjusted. The engines will then be in readiness to be started immediately in response to orders received by the engine-room telegraphs.

Precautions: (1) opening the manœuvring valve.—This should not be opened sufficiently wide to start the engines rapidly, as this might cause damage to the cylinders and pistons if the former should happen to be not properly drained, or cause sudden reduction of the pressure in the boilers, and possibly sudden ebullition and consequent passage of water directly from the boilers to the cylinders. The amount of opening required should, however, be quickly given. If the main regulating valve be used for starting, a very small amount of lift will be sufficient, and additional care is necessary to prevent too rapid starting.

(2) Shutting the steam supply-valve whilst reversing the links.—With flat slide-valves, and especially if the engine is at rest, this should always be done, as the initial friction of such slide-valves is considerable, and even if the starting gear is strong and sufficiently powerful it ralieves it of considerable stress. With many of the older starting gears, and with unlubricated flat slide-valves there is sometimes

risk of straining the gear. If the engine is moving, the stress on the gear is much less. Independently of this, however, most engines have a certain position from where they start less readily than from others, and if steam is on and the starting gear be slow in action, the gradual admission of steam after the links pass the central position often causes the engines to move into this unfavourable position, while if the steam is not admitted till the links are in full gear it permits of more direct control over the movements of the engine, as a greater volume of steam is admitted to the cylinder through the slide-valve.

This applies especially to the older compound engines, with flat high-pressure slide-valves, and with little margin of power in the reversing engines. With modern triple-expansion engines, after once under way, and when time of reversal becomes a consideration, this practice is not an absolute necessity, as the starting engines and reversing gear are sufficiently powerful to move the link gear over quickly enough without closing the steam supply valve. When cylindrical slide-valves are fitted, little additional work is brought on the starting engine and the closing of the steam supply valve is unnecessary.

(3) **Use of the auxiliary starting or pass-valves.**—These valves should be worked with judgment, or possibly their use may defeat the object in view, and prevent the engine from starting.

(*a*) With pass-valves. In triple-expansion engines with valves fitted so as to admit steam to the intermediate and low-pressure *receivers*, care must be taken in the slide-valve setting of the engines that the whole of the main slide-valves cannot be closed to the admission of steam at any position of the cranks. In handling the engines by means of these pass-valves, only that valve should be used which will be effective in producing a turning moment in the required direction of the shaft. For example, suppose that when the high- and low-pressure slide-valves are closed to steam, but the intermediate slide-valve is open, that the pass-valves leading to both receivers are opened. The effect will be to put steam on the proper side of the intermediate piston from the intermediate receiver, whilst owing to steam having been admitted to the low-pressure receiver, it finds its way through the intermediate exhaust to the other side of the intermediate piston, thus tending to neutralise the effect of the other pass-valve, at the same time the other two cylinders are of no use for starting purposes, the result being that the engine may not start. If steam is admitted only to the low-pressure receiver, besides producing no effect in starting, it is also objectionable for the reason that the exhaust pressure tends to force a flat intermediate slide-valve off its face.

The correct auxiliary valve to open in the above case is obviously that to the intermediate receiver only, when the turning force available for moving the engine is due to the pressure on the intermediate piston less any adverse force due to the same pressure acting through the exhaust on the high-pressure piston. With the above fittings the turning force available can generally only be due to the receiver pressure acting on the difference of areas of two adjacent cylinders, and as the steam has also to fill the whole of the receiver, and generally portions of two cylinders—viz. the steam side of one and the exhaust part of the preceding—they are sometimes slow in their action. For this reason starting valves are by some preferred to be fitted to supply

steam directly and independently to each end of each of the intermediate and low-pressure cylinders.

(*b*) When such starting valves are fitted, the force available will be due to the starting pressure acting at least on the area of one of these cylinders, and frequently may be due to that pressure acting on the sum of the areas of the low and intermediate-pressure cylinders. Owing to the smaller space to be filled with steam, their action is, when correctly used, more rapid than the preceding variety. With these fittings, however, more care is required to admit the steam to those ends of the cylinders which will cause the motion of the pistons to rotate the cranks in the direction corresponding to the position of the links, or similar complications to those previously described will occur, besides which, in the case of flat slide-valves, there is a liability to force these valves off their faces; more consideration as to the side of the piston to which it is necessary to admit steam is required than with pass-valves, so that the latter are the more simple in use.

The auxiliary starting valves should therefore be used as little as possible, and then with care as to direction. When an engine becomes locked by the injudicious use of the starting valves, the cylinders should be freed from the steam as quickly as possible, and this is most readily accomplished by closing the starting valves and main regulating or manœuvring valves, and putting the links into full gear in the opposite direction, at the same time opening the cylinder and slide-casing drains, thus permitting the steam in the engines to pass through the exhaust to condenser and bilge as necessary. The engine should then be moved in this direction, after which the link-gear should be quickly reversed, when, if mistakes are not repeated, the engines will start.

Most of the earlier compound engines were fitted with flat slide-valves on the high-pressure cylinder, and these are with inexperienced operators liable to be forced from their faces when using the starting valves. This is shown by a sudden rise of pressure in the L.P. receiver, which may cause the relief valve to lift and steam to escape to the engine room. The regulating or manœuvring valve and starting valves should then be closed, the link-gear worked to and from its mid position in both directions, to relieve the low-pressure receiver from pressure, and the cylinder and receiver drains opened to relieve the pressure in the cylinders, after which the link gear should be middled and the regulating valve suddenly opened, and the reversing engine worked so as to move the link-gear a short distance to and from the mid positions in both directions, when the slide-valve will usually assume its normal position. If the pistons are leaky difficulty may arise in starting the engines with the auxiliary starting or pass valves due to the steam leaking past the piston and thus escaping to the exhaust.

(4) **Warming cylinders.**—If this is not attended to, difficulty will be experienced in starting owing to the considerable condensation of the entering steam, and the consequent accumulation of water will be conducive to damage to the cylinders and pistons, unless great attention be paid to the drainage arrangements. Abnormal stresses are also brought on the cylinder castings by expansion through the sudden admission of steam to comparatively cold surfaces.

(5) **Draining cylinders.**—Previously to trying the main engines, the cylinders and slide casings should be blown through by the use of

the starting valves, &c. This permits any excess of oil present to pass direct to the bilge. When under way these directing cocks should be turned to discharge to the condensers.

In vessels fitted with horizontal engines in which the main condenser is higher than the lowest part of the low-pressure cylinders, difficulty has frequently been experienced in preventing accumulation of water in the low-pressure cylinders, especially when the vessel is under way, as the difference of vacuum in the condenser and low-pressure cylinders is not sufficient to overcome the resistance of the drain pipes and to lift the water from the cylinder to the condenser. To meet this difficulty small evaporators have been fitted below the lowest parts of the cylinder, and the cylinder drain pipes led to them, a live steam coil being used to evaporate the drain water. The usual drainage arrangements to bilge are also provided.

(6) Accumulation of fresh water in condensers before starting.—It is of importance to prevent this accumulation, especially when the main air-pumps are of long stroke and consequently have a large bucket velocity. This is now usually effected by a suction from either the auxiliary air- or hot-well pump, which pump must be kept working. If this be not attended to, the main air-pump may become glutted with water on starting, and great stresses be suddenly brought on all parts of the air-pump. It also usually entails a loss of fresh water through the air-pump relief arrangements. The water accumulates from steam and water passing through the engines during the warming process, and from the use of the silent blow-off. Accumulation may also be due to leaky condenser tubes or joints of tube plates, and this should always be ascertained whilst the engines are being prepared for starting by drawing off some and testing it. If leaking to any extent, it is advisable, if time be available, in order to prevent the access of salt water to the boilers, that the defect be made good before getting under way; if not, the first opportunity should be taken to effect the repair.

(7) Condenser pumps.—The main circulating pumps should be kept running sufficiently fast to insure the condensers being cool, and any drain cocks or other fittings which may admit air to the main condensers should be adjusted to prevent this. The auxiliary circulating pump and air-pump should also receive attention, so that the auxiliary condenser remains cool and free from water, which insures efficient drainage from those cylinders and slide casings led to the auxiliary exhaust pipe and from the cylinder jackets. It also relieves back-pressure from the auxiliary engines generally.

(8) Safety valves blowing off or steam pressure rising too high.—This should be prevented to avoid the loss of fresh water, and inconvenience from noise and water on deck. If the steam pressure tends to rise faster than required, the fires should be checked as follows. The stokehold fans should be run at a speed only sufficient for ventilating purposes, while the draught plates on the boilers should be closed, and the fires should not be disturbed more than actually necessary. If the pressure still continue to rise beyond that required, the silent blow-off valve should be slightly opened so as to pass the steam direct to the main condenser. This should always be done with caution, and care taken that (*a*) the valve is opened gradually, so as not to shake the condenser tubes more than necessary, otherwise leaky tubes may result,

(*b*) the pump for emptying the condenser of fresh water is at work, (*c*) the silent blow-off is shut before the main engines are started. The steam pressure should be checked in this manner when easing or stopping the engines.

Starting of engines deferred.—In cases where some considerable time elapses between the first trying of the engines and the order to start being received on the engine-room telegraph, in addition to the preceding, the starting engine should be worked occasionally from full gear ahead to full gear astern, so as to prevent accumulation of pressure in the receivers and keep the working parts free.

Procedure when main engines are under way.—When the engines are under way the suction from the auxiliary air-pump or hot-well pump to the main condenser should be shut, and the speed of the main circulating pumps be regulated to the requirement of the main condenser. The engine-room tell-tales should be taken out of gear, except in the case of warships steaming with a fleet, in which case they may still be required. Special attention should be paid to the bearings and all working rods to see that they are keeping cool, also to the engine-room gauges, to see that the vacuum is maintained in the condensers and that the jacket and receiver pressures are correct. The engines should be worked slowly for a time, and then the regulating valve slowly opened, so that the speed is gradually increased and no sudden demands are made on the boilers, which would result in a lowering of steam pressure and possibly passage of water with the steam into the cylinders. This also allows the fires to gradually burn through and attain a sufficient incandescent body for maintaining the required speed without undue forcing, and brings the maximum stresses uniformly and gradually on the various parts of the engines and boilers. Especial attention should be paid to the lubrication of the thrust-block at this period, as during the process of starting and till the ship is fairly under way the thrust on it is in excess of the normal amount.

Adjustment of link gear.—When fairly under way the link motions should be adjusted to the cut-off which will give the required number of revolutions of the engines, with the regulating valve practically wide open, to obtain in the high-pressure slide-chest a high steam pressure, in order to promote economy of fuel, unless the cut-off required would be too early for smooth and regular working, in which case the link motion should be set for the earliest practical cut-off and the steam pressure adjusted accordingly. This point occurs generally at or before 2-10 stroke, or at such rate as develops either (*a*) an excessive compression loop at the top corner of the indicator diagram, (*b*) too considerable a falling away of the high-pressure admission pressure due to wire-drawing, or (*c*) a jerky motion of the valve gear. The nature of the limit will depend on the design of the valve gear, and especially the amount of compression in the high-pressure cylinder at full power. With many valve gears, however, the jerky motion referred to, imposes the limit of efficient working. When this point is reached there is no alternative except to obtain further reductions of power by lowering the initial steam pressure.

Independent linking-up gear.—This should now be adjusted, depending on the speed required, so as to regulate the proportion of power developed in the several cylinders. Experiments should be made, as far

as possible, when making passages, with the view of determining the best distribution from the point of view of economy, the cut-off in the high-pressure cylinder being always as early as practicable. The best distribution for various rates having been thus determined, the engines can, when fairly under way, be at once properly set, if continuous working at that speed is expected.

Jacket pressures. Scrooping or grunting.—The cylinder jacket pressures should be regulated, and it should be borne in mind that the higher the pressures in the jackets the less will be the loss from 'exhaust waste' due to liquefaction and the consequent wasteful transfer of heat direct to the exhaust. The most efficient plan, considering only the action of steam, would be to carry steam of the full boiler pressure in each of the cylinder jackets, including the low-pressure, but even if this were done it would be insufficient to entirely prevent loss from this cause. Owing, however, to mechanical considerations connected with the wear of the internal parts, as the amount of internal lubrication in high-pressure marine engines is always reduced to a minimum, it is not possible to carry out the principle to this extent, especially in the low-pressure cylinder. The jacket pressures should generally not fall below the initial pressure in the cylinder—i.e. the preceding receiver or steam-pipe pressure—in which case the amount of moisture present in the cylinders will be considerable. In some cases where wear is feared the high-pressure steam jacket supply is taken from the high-pressure slide casing. The pressure which can be safely carried can only be determined by experience, and this will vary with the amount of lubrication entering the cylinder through the piston- and slide-rods, the material of the rubbing surfaces, and their adjustment.

If any 'scrooping' occurs, which can be made to disappear by lowering the jacket pressure, the limit will have been reached, but it should not be too hastily assumed that noises of this kind are caused by the jackets. Scrooping often occurs when the speed of the engines is reduced by throttling the steam, which wire-draws and dries it. The boiler pressure should therefore be adjusted to the requirements of the engines, provided it is not reduced below that required for readily starting from any position.

Feed-water temperature.—Where the circulating pumps are fitted in duplicate, as usual in the Navy, only one should be worked, and the speed of this should be reduced until the temperature of the feed-water is as desired. With feed-water heaters there is no limit to the lowness of this temperature except that of efficiency. Without them the effect on the boiler has to be considered, and under these circumstances it is not advisable to reduce this temperature below 100° F., so as not to unduly magnify the racking strains on the boiler due to differences of temperature. With feed-water heaters the temperature may be rather lower with advantage. When working at low power it will often be found that it is impossible to run the circulating pump so slowly as would be required from these considerations without risk of its stopping altogether. If this be so, in order to avoid this danger, and at the same time where feed-water heaters are not fitted, to prevent the feed-water being too cold, the circulating water valve should be gradually closed until the feed-water is of the correct temperature, with the pump running at a speed which will avoid danger of stopping.

constantly observed. If automatic float gear is not fitted, the speed of the feed-pumps should be regulated so that the suction is always covered with water to avoid air entering the pumps and the boilers. If additional feed-water is required beyond that which the evaporators are producing, it should be obtained from the reserve fresh-water tanks by opening the suction from this part to the main condenser or air-pump.

If when the engines are working steadily any considerable discharge takes place from the overflow-pipe on the feed-tank, the engine-room watchkeeper should at once warn the stokehold staff, and ascertain the cause, which may be due to the feed-pumps stopping or failing to deliver sufficient water to the boilers, or to the boiler feed-valves not being sufficiently open. Prompt action under these circumstances has prevented many accidents.

CHAPTER XXX.

MANAGEMENT OF ENGINES UNDER WAY—ENGINE AND BOILER DEFECTS.

In this chapter, and the succeeding, particulars are given relative to the care and management of the machinery and boilers, which are to some extent based upon the instructions contained in the 'Steam Manual' now in use in the Royal Navy. A few of the most usual defects in engines and boilers are also dealt with, and suggestions given as to their suitable treatment.

Before taking charge of a watch the engineer should satisfy himself by personal inspection that all bearings are working well and are in a proper state of lubrication, that the water in the boilers is at the proper working height, that the water in the bilge is not excessive, and that as far as can be seen there are no recent defects of importance in machinery or boilers. He should read the recent entries in the engine-room register, and make himself acquainted with any special orders. Pending these inspections the engineer about to be relieved remains in charge. He should also receive information from the off-going engineer, showing that no increase of density in the boilers has taken place, or, if the water has become salt, that it does not exceed the density allowed, and also as to what special cocks and valves are open in the department. The information as to these points should be verified by the oncoming engineer as soon as possible after taking charge.

When in charge of the watch he should leave the engine-room platform as little as possible, so as to execute expeditiously any order received from the deck or stop the engines in case of accident. As his duties require his occasional absence from the engine-room platform, he should instruct one of the subordinate watchkeepers in the manipulation of the engines, and this watchkeeper should be always on the starting platform during his occasional absence.

In addition to any special orders received, the watch has to carry out the usual routine orders, and note their execution in the engine-room register. The principal are—

Hourly.—Quantity of coal used; steam pressure in the boilers and at engines; vacuum in main condensers; air pressure in stokeholds, if any; total number of revolutions; depth of water in the bilges.

At least each watch.—Density of the feed-water and of the water in each of the boilers in use; temperatures of feed-water, engine and boiler rooms, coal-bunkers, on deck, and sea-water; quantity of fresh water made for drinking purposes and used for make-up feed (measured

description of any work done in the department.

Other notations in register.—The other necessary notations of work done to be entered in the register beyond that occurring each watch are as follows :—

Daily.—The alkalinity of the boilers in use ; temperature of bunkers when fires are not alight. Take indicator diagrams and insert horse-power, &c., which should be done more frequently should any material variation occur in the working condition of the engines. Moving of any auxiliary engines not in use ; working the siren and whistle ; examination of controlling shafting of the steering engine, and statement of condition.

Four times per week.—Removing bunker lids for ventilation purposes where bunkers are not automatically ventilated, and replacing them after three hours. Even where automatic ventilation is fitted it is desirable as a precautionary measure to frequently remove the lids.

Weekly.—Lifting safety valves of, and testing density and alkalinity of water, in boilers not in use ; working the various cocks and valves, especially those of sea connections ; condition of all telegraph gear ; working all watertight doors, sluice, drain, and ventilating valves.

Use of oil.—Each watch is allowed certain fixed quantities of the various oils, which have been ascertained by experience to be sufficient for the good working of the machinery. Every endeavour should be made to keep within the quantities allowed. The principal used are olive oil, for external bearings ; or, in cases *where the load is not heavy*, a mixture of olive and mineral oils is used for this purpose. Mineral oil is used for internal lubrication and for piston, slide, or fresh-water pump-rods, and on any other fittings where it can get mixed with the feed-water. Colza or rapeseed oil is used for lamps.

Cleanliness.—This is essential for health and the efficiency of the machinery department. In a well-ordered engine room, each watch, in addition to the general duties, is responsible for a definite portion of the cleaning work. During long steaming periods, especially in tropical climates, the bilges require constant attention ; and those parts which are not easily accessible under way should, whenever practicable, be frequently washed through, and be freely supplied with disinfectants.

Density of water in the boilers.—It is the usual practice, when under steam, to test the water in the boilers by the hydrometer every watch ; and when not under steam, once each week. As a rule, if sea-water necessarily has access to the boilers, no brining or emptying is necessary until the density reaches 25 degrees, or 2½ times the density of sea-water, but it should not be allowed to exceed 40 degrees. The selection of the most suitable density in any particular case must depend on the experience gained principally as to the condition of the boilers on the examinations after steaming. With surface condensation, if only salt-water make-up feed is available, the rise of density should not exceed one degree daily, a daily increase of two degrees being considered excessive.

In the more recent vessels, in which fresh water, either from the reserve tanks or from the evaporators, should always be available to make up losses, practically no rise of density should take place, and a rise of two degrees in three days is considered excessive. When an

excessive rise of density does occur, steps should be taken to ascertain the cause and to remedy the defect. It is now very rarely that the density in such vessels rises to the above given limits, and it is generally only necessary under way to get rid of the grease or other impurities from the surface by opening the brine cocks occasionally. The deposition of scale in the boiler does not depend so much on the density at which the water is kept as on the quantity of sea-water that enters. The higher density has, in the case of boilers fed from surface condensers with salt-water make-up, been found beneficial as regards cleanliness as well as economy.

Change of water.—On service, in order to exclude air, in the absence of which oxidation is small, it is desirable to keep the water in the boilers without change as long as practicable, whether the fires are alight or not, the boilers being only emptied when necessary for examination, cleaning, or repair. This will require to be done much oftener with boilers fed with sea-water than with those fed from surface condensers, as in the former case scale is continually being deposited on the heating surfaces while the boilers are at work. Experience as to the condition of the boilers on the examinations can be the only guide as to the greatest amount of steaming to which it is safe to expose them without cleaning. This depends principally on the amounts of solid and greasy matter admitted with the feed-water.

When filled with fresh water, with the condenser tubes tight, and sea-water make-up, it has been found that on an average, boilers should be cleaned after twenty-five days' continuous steaming at moderate speeds. Intermittent steaming under the above circumstances tends to cause the scale to accumulate on the furnace crowns, in which case cleaning has been found necessary after fires have been alight sixteen days. The higher the density at which they are worked, the longer they can be kept going without cleaning.

When distilled water is used for make-up feed, and a moderate amount of oil is used for internal lubrication, no difficulty is found in steaming for thirty days continuously at moderate speeds without removing the greasy deposits from the boilers, but it depends entirely on the amount of oil used. In many cases, using little oil and with efficient grease filters, much longer periods than this can be run without opening up.

Chemical tests.—It is necessary that the water in the boiler should be prevented from becoming acid by decomposition of lubricants or other causes. If it be kept in a neutral or alkaline state it will possess few corrosive properties. In order to ascertain the chemical condition of the water, a small quantity should be drawn off every day when the boilers are at work, and once a week when not at work, and tested with litmus paper. If the water is acid, it may be neutralised by putting a little soda in the condenser or hot well, from whence it will be pumped into the boilers with the feed-water. The use of lime also has the effect of keeping the water in the boilers alkaline, and lime dissolved in a special tank and then admitted to the feed-water is used in preference to other such substances with Belleville boilers.

Uniformity of temperature and pressure.—Sudden changes of temperature and pressure during the working of the boilers should be prevented; the smoke-box doors should only be opened when abso

lutely necessary, as in tank boilers the cold air affects the tube ends, which, being thinner than the tube plates, are liable to shrink and leak; but this is of little importance with water-tube boilers.

The boilers should not be emptied by blowing out except in cases of urgency, as this produces strains, and consequently leaks, especially in tank boilers. When steam is no longer required, the boilers should be closed up, the fires allowed to burn out, and the water cool gradually, so that the boilers may contract uniformly, and prevent undue strains on any part. When all is cool, the ashes and clinkers, &c., may be drawn out of the furnaces, and any fuel not entirely consumed should be saved. The water is, when cool, run out from the boilers—if necessary for examination, cleaning, or repair—into the reserve fresh-water tanks if it be clean and fresh, or if otherwise to the bilge. With tank boilers, the practice of drawing fires shortly after the engines are stopped is objectionable, as the cold air entering the furnaces is liable to cause leakage at the fire-box ends of the tubes, and in the joints of the combustion chambers and backs of the furnaces.

Stoking and economy of fuel.—Care is required in the working of boilers to promote economy of coal. The fires should be stoked carefully and regularly, the steam pressure and water level kept constant, and no waste allowed by steam blowing off from the safety valves, which should always be kept tight to prevent leakage. The cinders and small coal that fall into the ashpits should be reburnt, and the fires not unnecessarily forced or disturbed. The wing firebars should be placed close to the sides of the furnaces, to prevent the fire being too fierce close to the plates. If there are stay-nuts or other projections in the line of fire-bars, the wing-bars should be cast with recesses to fit over them, so that they may be quite close to the plates. In corrugated furnaces the wing-bars should be cast to fit the corrugations.

To diminish the quantity of cold air admitted at one time, also to reduce the bad effects of this air striking the tube plates, no two furnace doors of the same boiler should be open at the same time, and the firing should be done quickly, the furnace doors only being kept open a short time. When using Welsh coal in ordinary tank boilers under natural draught, five to six inches is a good thickness of fire, which should be increased when working under forced draught. With Belleville boilers a rather thinner fire than this gives the best results. The fires in all boilers should be maintained in the same condition to equalise steam production, and kept as free from clinker and slag as possible to insure access of air through the bars.

The amount of clinker can be judged by the brightness or otherwise of the bars when viewed from below. In addition to the partial removal of clinker during every watch, each fire is consecutively thoroughly cleaned; for which purpose it is burned low, and all clinker and slag, with most of the fire, is removed from the furnace bars. During the cleaning the uptake damper of this furnace, if fitted, should be closed, and fan draught reduced to prevent much access of cold air. To reduce fluctuation of steam pressure only one fire is cleaned at the same time, and only one fire in the same boiler during the same watch, if practicable. When working under natural draught with Welsh coal, it is generally arranged that each furnace is cleaned once during

each twenty-four hours, or oftener with higher rates of combustion or with inferior coal.

It is important to keep the tubes and smoke-boxes free from soot and ashes. Soot doors are now generally fitted, so that this can be done as regards the smoke-boxes without opening the smoke-box doors. For the tubes, brushes are supplied for use when steam is not raised; but under way the cleaning is done by means of the steam-jet. Under natural draught this is usually carried out on each boiler once in forty-eight hours, and, owing to the discharge of small ashes, &c., on deck, is done at night during the first and middle watches. During the operations of cleaning fires or tubes, the tendency of the steam pressure is to fall, and despatch is therefore essential.

Losses of feed-water.—All losses should be reduced to a minimum, and a supply of fresh water to make up the losses should be provided. Losses occur through leakages from boilers, engine glands, and drains of various kinds, air-pump and other fresh-water pump-rods, defective pipe joints, lifting of safety and relief valves of boilers and engines, air-pump relief when engines are suddenly started, feed-tanks overflowing to bilge, using the siren and whistle, sweeping tubes by steam, scumming boilers, leaky cocks and valves, &c.

The above losses amount on an average to from $\frac{1}{4}$ ton of water for each ton of coal burnt when steaming at high powers, to $\frac{1}{2}$ ton of water per ton of coal at low powers, the increase being due to the losses by auxiliary machinery being comparatively large and practically constant at all powers. Losses of feed-water are replenished either from fresh water carried in the reserve tanks, or by using the evaporators in connection with one of the auxiliary condensers.

Use of oil for internal lubrication.— In modern high-pressure marine boilers the principal danger to be guarded against is the mineral oil deposit on the furnaces and combustion chambers. The oil used for the lubrication of the cylinders and slide-valves is carried over into the boilers, and sometimes forms a black carbonaceous scale, which is an almost perfect non-conductor of heat, even when not more than $\frac{1}{16}$-inch thick. In order to prevent as far as practicable access of oil, all internal lubrication of the main and auxiliary machinery should be reduced to a minimum, and oil on piston-, slide-, air-, and feed-pump rods should not exceed that necessary for good working. It is generally found unnecessary to admit lubricant directly to the internal parts of the engines, as that which enters the cylinders by means of the piston- and slide-rods is found to give sufficient lubrication for the internal parts. All grease extractors and filters should be kept in working order, and frequently cleaned.

Feed-water increasing in quantity.—If without the use of the evaporators for make-up or of additional feed, the water in the boilers shows little or no loss, or should be gaining instead of losing, the reason should be ascertained. Test the feed-water, and if there are no indications of salt, the gained water is fresh, so that the *make-up* feed-valves on the suction to reserve fresh-water tanks should be examined to see that they are not open or leaking, or that the auxiliary feed-pump suction to the reserve fresh-water tanks is properly closed.

Should the feed-water be salt to any degree, ascertain the cause, and remedy it at the first opportunity. Saltness is occasioned princi-

pally by defective condenser tubes or tube glands, stay-nuts, tube-plate joints, &c., leakage through the sea connections of the auxiliary feed-pumps, or priming of the evaporator, and there are other minor causes.

Fan engine stopping.—Should this occur when under air pressure, the damper in the fan trunk should be closed to prevent loss of air pressure and the speed of the remaining fans correspondingly increased. The cause, which is generally defective lubrication, should be remedied and the fan re-started as soon as possible.

Flaming at the funnel.—If this occur see that the cowls on deck are trimmed to the wind and the fans are running sufficiently fast; the fires should be levelled and reduced in thickness, and where the furnace doors are so arranged, as much air as practicable should be admitted over the top of the fire. It is generally due to unconsumed gas—i.e. carbonic oxide—bursting into flame on meeting the oxygen of the air at the top of the funnel. Reducing the thickness of the fire admits more oxygen for combustion. It may even be necessary, in torpedo operations at night, when the boilers are not being pressed and concealment is desirable, to slightly open the furnace doors to prevent flaming.

Furnace bars melting.—Any such bars should be replaced by removing the fire from the part affected, and checking the air supply to that furnace by closing the damper and draught-plates, also easing the fans. The bars are generally slid into their places by means of fire-irons, or by tying the bar to a slice, and dropping it in place. The cause will often be found to be the absence of water from the ash-pans or fires being unduly thick.

Priming.—The passage of water with the steam from the boilers to the engines, technically called *priming*, is most objectionable, and when it is severe, unless proper precautions be taken, may lead to dangerous consequences, such as the cracking of cylinders or covers or canting of pistons by blows due to the thumping of the pistons at each end of the stroke.

Priming arises either from unduly forcing a boiler, more especially if the design be defective, or from bad management.

Points in design affecting priming.—As regards design, this should provide for the proper circulation of water, absence of local ebullition, and the free escape of the steam generated; otherwise there will be danger of priming when the boiler is being forced. These points are favoured by:—

(*a*) *Ample water spaces* round the sides and backs of the combustion chambers and between the tubes. In large boilers, the clear space between the tubes should never be less than one inch, and this might be increased with advantage in a horizontal direction if space and weight are available; also, the lower rows should be kept well clear of the tops of the furnaces.

(*b*) *Volume of the steam space being sufficiently large.*—If the space allowed provides too small a reserve for steam, fluctuations of pressure are caused, and irregular ebullition, which may cause water to pass over with the steam. The volume per maximum horse-power with cylindrical boilers should not be less than from ·5 cub. ft. at 60 lbs., to ·2 to ·25 cub. ft. at 155 lbs. pressure, and more should be allowed if possible. The limit to which the volume may be reduced depends on the type of boiler, for in torpedo-boat destroyers this value has been

reduced to 65 and 60 cub. ft., with pressures varying from 180 lbs. to 200 lbs. per sq. in., without any detrimental effects due to priming, but in these cases the directions of the steam and water currents have been especially provided for.

(c) *Sufficient area of the water surface.*—The larger this area the less is the quantity of steam per unit of surface which passes from it, and consequently the more gradual is the escape of the steam.

(d) *Suitably arranged internal steam pipes.*—These should, where practicable, be arranged so as to collect the steam uniformly from all parts of the boiler, and be placed as far as practicable from the water level. They are often arranged as two complete pipes running the entire length of the boiler, and are provided either with small holes or slots on their upper surfaces. The total area of these orifices should not be less than twice the area of the pipe leading from the boiler.

Bad management as causing priming—

(a) *Starting the engine or increasing speed too suddenly.*—This is a most injudicious practice, as it rapidly lessens the pressure of the steam in the boiler, and consequently the corresponding temperature. The temperature of the water will therefore be in excess, and violent ebullition and possibly priming follows.

(b) *Too much water in the boiler.*—This produces small steam space and water surface, with their consequent bad effects.

(c) *Uneven firing*, or uneven forcing of different boilers. Uneven firing causes different rates of combustion with various furnaces, interfering with the normal circulation and producing irregular local ebullition. Boilers nearest the engines, or having a freer supply of air to the furnaces, frequently do more work than others in a set, unless the supply from them is regulated, which is usually done by partially closing the stop-valves from those boilers.

(d) *Impurities in the water*, such as the presence of too much soda, from not having sufficiently washed out the boilers after having used soda for cleaning them, or from the use of soda in cleaning the condenser tubes, causes priming. In many types of boilers it is certain to result in priming, even at moderate speeds. Boilers of the locomotive type are especially sensitive to this action, certain qualities of fresh water from the shore even setting up a strong tendency to frothing and priming. Mud or excess oil will also cause priming. They do this in various ways, sometimes presenting a resistance at the water level to the passage of the steam generated, causing local, intermittent, and violent actions.

To stop priming.—If it be general in all boilers, reduce the speed of the engines and decrease the rate of combustion by easing the fans or shipping the draught plates, care being taken to keep sufficient water in the boilers, the auxiliary feed system being used if necessary.

If only one or two boilers in a set are priming, the cause will be local and be generally due to improper management. In this case the stop-valves on these boilers should be partially closed and the rate of combustion checked. If due to management, either (a), (b), or (c), as soon as the water is quiescent and at its working level, the stop-valve should be gradually opened and the boilers gradually worked up to the required power.

If all the boilers prime, due to (d) impurities in the water, the

clean fresh water from the reserve feed-tanks and distilling plant till the water in the boilers is mostly renewed, together with the use of the surface blow-off to remove impurities, is beneficial. Formerly, when sea-water was generally used, the same result was obtained by keeping the surface blow-out open and using sufficient salt-water feed. If the cause is the presence of soda, the only cure is to have the boilers emptied and washed out.

If, after all precautions have been taken, and with good working conditions, it is found that on the engine reaching a certain speed the boilers still prime, it may be concluded that they are being forced beyond their immediate capabilities, and the generation of steam will have to be reduced for safe working.

Indications of priming.—1st. By the water in the gauge glasses of the boilers being considerably agitated and not showing a constant level, broken water often passing continuously through the glass. If due to dirt in the boilers, the water in the gauge-glass becomes discoloured.

2nd. The separator, if fitted, gets filled with water, and its drain should be kept open to prevent as much water as possible from passing to the engines. If a separator be not fitted, the drains on the valve-boxes in the line of steam pipe should be kept open for this reason.

3rd. Water comes over into the slide casings and cylinders, and causes a knocking at the ends of the stroke. If the priming become excessive this causes the cylinder relief valves to lift, and may be dangerous if too great to be relieved quickly. The drain cocks on the cylinders and slides should be kept open and the speed of the engines reduced.

4th. Even slight priming causes the speed of the main and other engines to be reduced, which reduces also the speed of the air-pump, while, as the quantity of steam and water passing into the condenser is increased, the vacuum in the condenser becomes reduced, often considerably. As the water carried over with the steam evaporates during exhaust, the back pressure in the cylinder is increased. The air-pumps also may be overcharged and the feed-tanks quickly filled.

Broken gauge-glass.—The cocks on boiler should be immediately shut, and the glass replaced without delay. The new glass should be free from flaws or scratches, with ends ground square or fire-finished, and of the correct length. If too long it would restrict the passage of steam to the glass, if too short the packing may work over the edges of the glass. In the later Admiralty pattern gauges the correct length of the glass is stamped on the framing. Before replacing it the whole of the old packing should be removed, and the screws of the adjusting glands made easily workable.

The packing is then placed on the glass, and while screwing up the bottom gland, the glass should be kept in contact with the metal of the lower cock. Care should be taken not to screw up the upper gland hard before the lower one, as this may lift the glass and perhaps allow the lower packing to choke the orifice. The glands should at first be only screwed up hand-tight, after which the *steam* and *drain* cocks should be opened a small amount to heat the glass gradually, when after a short interval the *water* cock should be gradually opened, and then

the drain cock closed, the steam and water cocks being then fully opened gradually, and the glands adjusted as required. These precautions are necessary to prevent fracture of the glass.

The glass should now be tested by closing the steam cock and opening the drain, when water should rush out freely. Next close the water cock and open the steam cock, when steam should freely rush out. Next close the drain and open the water cock, and *note carefully how the water rises in the glass.* It should rise smartly to the water level of the other glass.

Draining water gauges.—When, in the ordinary course, gauge-glasses are blown through or drained, by opening the drain cock, if the water rises slowly, but eventually stops at the same level as in the other glass of the boiler, this will indicate a partial choking of the lower passage. The choking of the upper or steam passage is an occurrence which very seldom happens. It would be indicated, however, by a rapid rise of the water level to a higher position than shown on the other gauge, due to condensation of the vapour in the upper part of the glass when cut off from the boiler.

To endeavour to clear glasses with such indications, blow through the steam and water cocks separately, shutting one before blowing through the other ; but if this be not successful, then the passages will require to be cleared by removing the screwed plug, and passing a small rod through the passages, removing the glass if necessary, steam being lowered in the boiler.

When the water gauges are not fitted directly on the boilers, but either on short stand-pipes or steady pipes, there will be a connecting passage between the upper and lower arms, in addition to that through the water gauge, and this has an important influence when dealing with defects in gauges, which must be carefully borne in mind. Suppose, for example, that the orifice in the stand-pipe leading to the steam space is choked (refer to Fig. 93); if, when testing such a gauge, the water cock be shut and the steam cock blown through, water would be blown out, ascending the stand-pipe from the lower orifice, and unless the presence of the stand-pipe is remembered, this might lead to the erroneous conclusion that the passages were clear. Under these circumstances the glass would be practically full of water, so that it would be wrongly concluded that there was too much water in the boiler.

For this reason it is usually insisted that the short stand-pipes fitted in the Navy (see Fig. 93) shall not be less than two inches in the bore, so that there will practically be little or no risk of their choking. With the long stand-pipes fitted in the mercantile marine (Fig. 94), cocks are fitted at each end of the stand-pipe. When these cocks are fitted the orifices of the stand-pipe can also be tested, for by shutting the top one and blowing through the bottom, and *vice-versa*, it can be ascertained if they are both clear. The complete testing of such a gauge-glass and fittings, therefore, involves four distinct operations—viz. the independent testing of top and bottom orifices of the gauge-glass, and the same for those of the stand-pipe.

To ascertain whether a glass is full of water.—When filled with water well above the top cock, the water runs through the glass without a break when using the drain only, and if the water be clean no

movement will be seen through the glass, and it is sometimes difficult to tell whether it be full or empty. In this case shut the top and bottom cocks and open the drain, when, if the glass be full, the water will be seen gradually falling in the glass. Next shut the drain and open the bottom cock, keeping the top cock closed, when the water will be seen rising in the glass.

Water out of the gauge-glass.—The action required will depend on the cause and the time since the water was last seen in the gauge-glass; it will require judgment to determine whether it is advisable to put on all the feed available or to draw fires. If the stokehold watch-keeper is reliable, and is sure that the loss was due to an irregularity in feeding, and that the water-level could not possibly be much below the bottom of the glass, also that the feeding arrangements are in good order, then all available feed should be concentrated on that boiler, including the auxiliary feed service, till the water appears in the glass, when the ordinary feed arrangements should be again used.

Although not visible in the glass, further information can be obtained as to the lowness of the water by closing the top cock, when, if the lower orifice in the stand-pipe or boiler is covered with water, it will immediately rise in the glass, owing to condensation of steam in the upper part. This practice should, however, only be resorted to under such circumstances, and will then be some guide to the person dealing with the defect as to the nature of the further steps required, having in view the position of this lower orifice in relation to the highest parts of the heating surfaces.

If, however, the boiler has lost its water through failure of the feed-valves on the boiler, or the feed-pumps to act efficiently, excessive leakage, or excessive priming, the following steps would usually be taken.

(*a*) The draught plates should be shipped, to prevent as far as possible any leakage passing into the stokehold.

(*b*) The fires should be drawn, or extinguished by the drenching apparatus, if fitted.

(*c*) All steam stop-valves on the boiler should be closed.

(*d*) When the fires are practically out, the safety valves might be slightly lifted to reduce the pressure, if in excess.

(*e*) If there be leakage into the combustion chamber or furnace, the fans should be kept running to insure the vapour passing to the funnel rather than into the stokehold.

If the feed-pump be working and not increasing the water in the boilers, the water supply arrangements should be examined to see that only the proper valves are open, and that there is water in the tank from which it is drawing. Leaky or split boiler tubes sometimes account for the persistent failure of the feed-pumps to maintain the water level. If the pump is working very rapidly the cause will generally be shortness or absence of the supply of water. To ascertain that the feed-valve on the boiler is working correctly the beat of the valve should be felt at each stroke, and the temperature of the feed-pipe near the boiler ascertained to be practically that of the feed-water.

This valve may not have been working efficiently and have allowed the passage of hot water back to the feed-pump. If jammed open, tapping with a hammer often frees it.

The passage of hot water from the boiler to the feed-pipe will affect the efficient working of the pump, as the delivery valves are seldom quite tight, which causes the pump casings to get hot, and consequently increases the elastic force of the vapour on the suction side of the plunger, and prevents the foot-valves from properly lifting, and decreases, or even entirely stops, the supply to the pump. In this case a bucket of cold water thrown over the pump barrels often causes the pump to start working. If the check-valve cannot be got to work correctly, the pump should be shut off and the auxiliary pump only used for supplying the boiler, and the valves examined and refitted as soon as practicable.

A feed-pump may also become inoperative through (*a*) the feed-water being too hot, when the action previously described occurs in the pump barrels. Beyond 130° Fahr. they cannot generally be relied on. (*b*) Accumulation of air in the passages between the suction and delivery valves, for which special outlet valves or cocks should be provided at the highest parts; or air may be admitted owing to the glands of the pump plungers not being tight enough.

Most accidents and breaks-down of pumps occur during the operation of starting them in a hurry, when they are probably imperfectly drained, &c.; and, considering the fact that the auxiliary feed-pump, when required at all, is generally wanted quickly, it is a good plan to keep both pumps slowly at work, when in case of any derangement of one the other can be increased in speed at once and without danger of accident.

Condensers.—The ends of the condenser tubes should be kept tight, new packings being fitted as required, to prevent mixture of sea-water with the feed. This is a point of importance, and the feed-water should be tested regularly to ascertain whether or not it is fresh. If the condenser tubes are leaking considerably, care should be taken to prevent water passing to the cylinders when the engines are standing, and the ends of the defective tubes should be repacked on the first convenient opportunity. The tubes should be frequently examined and tested to ascertain if any are perforated, and they should be kept as clean as possible to preserve their efficiency.

Defective vacuum.—The effect of priming in reducing the condenser vacuum has already been referred to. The most usual causes of defective vacuum are as follow:—

1. Insufficient supply of cooling water through the condenser tubes, causing only a partial condensation of the vapour.

This may occur through (*a*) condenser sea-valves becoming partially closed, or the passages or ends of the tubes becoming choked. In small ships of shallow draught, *weed-traps* are frequently fitted between the inlet valve and the circulating pump, and should be occasionally cleared. Choking of the grating of the inlet valve most frequently occurs with the auxiliary condenser. In later designs arrangements have been fitted for permitting one of the reciprocating pumps to discharge through this orifice with a view of clearing it; but if not, a diver is sent down. When the ends of the tubes are choked, the obstructions may be removed through the hand-holes on the covers after emptying the condenser.

(*b*) If the vacuum suddenly diminishes, the circulating pump may

should be immediately sought and removed. If due to any cause not immediately removable, such as hot bearings or other defect in the mechanism, it will be necessary to stop the circulating engine to cool or adjust it. Meanwhile, the other pump should be started, or the valve connecting the supply from the other engine room should be opened. If this cannot be done rapidly, the main engines must be eased down or stopped.

2. **Dirty condenser tubes.**—When foul from grease on the steam side or muddy deposits on the water side, their heat-conducting power is considerably reduced. It may sometimes be advisable to clean the tubes by filling the steam side of the condenser with a hot solution of soda, and allowing it to remain for some time, care being taken that as little as possible of the solution is afterwards introduced into the boilers with the feed. In a condenser in which steam passes through the tubes, the grease may be cleaned by means of a hot solution of caustic potash or soda, passed through the tubes by means of a brush with a long wire handle. When the tubes are very dirty—especially in recent practice, where the area of cooling surface is much less than formerly—it may be necessary to remove the tubes from the condenser and thoroughly clean them.

3. **Defective action of the air-pump** due to (*a*) defective valves. If the pump is a horizontal double-acting one, either the head- or foot-valves defective will reduce the vacuum. In a vertical single-acting pump, as long as the bucket and head-valves or bucket and foot-valves are intact, the pump will work satisfactorily and the vacuum will be practically unimpaired, especially if the bucket and foot-valves are correct; but should the bucket valves be displaced, the effect in spoiling the vacuum soon becomes apparent, even when the head- and foot-valves are both intact.

Indiarubber valves become defective through contact with mineral oil or hot water, which renders them plastic and causes them to swell, when they may overlap or their working surface become indented into the grating. When this occurs, if they are not too defective, the remedy is to trim the valves, and, if necessary, reverse them. Indiarubber valves will last longer if they are occasionally soaked in a solution ot soda. Metal valves last much longer than indiarubber, and vessels frequently serve a three years' commission without disturbing or replacing any of the original metallic air-pump valves. When they do become defective, it is through distortion or breakage.

(*b*) Air-pump plunger too tightly packed. This causes the barrel to work warm, and consequently raises the temperature and pressure of the vapour, hence reducing the vacuum. Pending a convenient time for the refitting of the packing, the pump should be worked with as much water as can be afforded, so as to cool the barrel, the extra supply being obtained from the reserve tanks, that not required for the boilers being again returned to the tanks.

(*c*) Leak in the air-pump rod glands. This practically only applies to horizontal air-pumps, for with vertical bucket pumps this gland is not exposed to a vacuum. It is, however, a most effective cause of reduction of vacuum in a horizontal double-acting pump, where the gland is in direct communication with the space between head- and foot-valves.

(*d*) Leaky plunger. This impairs the exhaustion of the pump, and hence a higher pressure in the condenser is required to lift the foot-valves. In a vertical single-acting pump the effect produced is the same as when the bucket valves are slightly deranged.

(*e*) If the pump be badly designed, either as regards size or more generally in regard to excessive clearance volumes at the ends of the stroke, a poor vacuum will result.

4. **Air leaks direct into the condenser,** which may occur through glands of main piston-rods, and slide-rods, and air-pump rods when worked directly off the piston, not being properly adjusted, and especially the low-pressure cylinder relief valves, drain cocks, or other fittings and joints, being either off their seatings due to dirt, &c., open to the atmosphere, or having slack glands. The expansion joints on receiver and eduction pipes not being properly packed or adjusted are frequent sources of loss of vacuum. One of the various fittings on the condenser itself may be left open or not properly adjusted, especially those out of sight, such as the drain cock on the air-pump suction pipe, and the connection to reserve fresh-water tanks, especially when these are empty. Leaks from glands and drains of any auxiliary engines and pipe connections in communication with the main condenser are common sources of loss of vacuum, especially if the engines are not at work—*e.g.* those from starting engines, feed engines, and additional main circulating pumps, where so fitted.

5. **A defective gauge** occasionally leads to erroneous conclusions as regards the vacuum. This defect can generally be detected by comparison of the indications of the compound and vacuum gauges on each condenser. The cock on the pipe leading to the gauge will sometimes be partially closed, or the gauge connections to the pipe leaky. A check on the gauges can be obtained from an indicator diagram, and comparing the back pressure line with the vacuum on gauge.

Hot piston-rods.—A rise of temperature in piston- and slide-rods above that of the normal working conditions is generally accompanied by a smell of burning oil; and if the rod cannot be felt, heating can always be confirmed by syringing a few drops of fresh water on the rod, when, if *hot*, the water will hiss and run off in a spheroidal condition, and, if very hot, oil applied in a similar manner will cause a dense smoke. If not very much above its normal temperature—that is, *a warm rod*—the gland which contains the soft packing should be slackened, and the rod be well supplied with mineral oil and a little fresh water from a syringe, the speed of the engines being reduced slightly if considered necessary. The use of tallow is very effective as far as cooling the rod is concerned, but should not be used, as it gets introduced into the feed-water and thence to the boiler.

When the rod is much heated—that is, *a hot rod*—the gland should be slacked as before, and the load removed from the rod to prevent it bending, for which purpose, and also to give it a greater chance of cooling, the engines should be considerably reduced in speed, or stopped if considered necessary. The rod should first be cooled by oil only, applied with caution. Never put the water service on a *hot* rod, as it is seldom heated uniformly throughout the circumference, and sudden cooling of one part more than the other is liable to cause distortion and bending, more especially with hollow rods. Piston-rods

always require especial attention, for, with metallic packing, hot rods may cause a serious breakdown and delay owing to the melting of the soft metals used in the packing, permitting them to run, and often cut the rods. The principal causes of hot piston-rods are as follow :—

1. **Glands not properly packed.**—Where asbestos or Tuck's packing only is used, care is required to insure that the packing is not ragged, too much worn, or too hard, that the proper size and number of turns are used, that the joints of each turn are broken so as not to be in line, and that the gland is screwed up squarely.

With metallic packing as described in Chapter XIX., if properly fitted, no abnormal stress can be brought on the rod, as the springs are so adjusted that when the box is jointed no more pressure is brought on the rod than practice has proved can be done with immunity from heating. In this case the gland with the soft packing is only of secondary importance, but care should still be taken to see that it is packed squarely.

2. **Neck-bush and gland too tight** on the rod, causing unnecessary friction.

3. **Piston-rod not working centrally** in the stuffing-box gland and neck-bush. In vertical engines this may be due to

(*a*) Wear of the crosshead guide not being properly lined up ;

(*b*) Gudgeon-pin brasses wearing unequally in a vertical direction, more especially where the top end of the connecting rod has two bearings.

(*c*) Crank shaft wearing bodily forward due to thrust-block not being properly adjusted.

(*d*) If cylinders are bolted together, proper allowance may not have been made when adjusting the various bearings to permit of the rod being central when the cylinders are hot.

(*e*) There is also with horizontal engines the wear of the pistons downward, due to their weight.

4. **Insufficient lubrication.**—The piston-rods are continuously lubricated by pipes led to holes at the stuffing-box, as shown in Chapter XIX, but they are so important as regards good working of the engine that this should not be entirely depended on, but be assisted with an occasional application of mineral oil from a syringe.

Causes of hot bearings :—

1. Deficient surface. The pressure between the working surfaces thus becomes so great as to prevent a film of the lubricant passing between them.

2. Working surfaces not being in line, causing abnormal local pressures and abrasion.

3. Bad fitting, so that the pressure is not uniformly distributed over the working surfaces.

4. Stopping of lubrication, which is the most usual cause.

5. Presence of gritty substances in the bearings.

6. Lubrication improperly applied. As a general rule the oil should be led to the bearing at a point of low pressure. If led near the point of greatest pressure suitable oil ways should be cut to allow the oil to enter the bearing.

When feeling a bearing to ascertain its temperature, care should

be taken to feel the actual bearing itself as close as possible to the rubbing surfaces.

Cooling hot bearings.—The following are general principles:—

(*a*) Where the heating is known to have been gradual, and the bearing is *warm* only. In this case supplement the oil supply with the water service. If the temperature still increases the bearing should be slightly slackened, or otherwise adjusted, according to the description of bearing. With main bearings, thrust and plummer blocks, stern glands, and many others, this can be done without stopping, but for gudgeon pins, crank-heads, &c., the engines must be stopped. When stopped, the bearing should be thoroughly cooled before starting again, and in the cases of crank-head and gudgeon pins care must be taken not to slacken sufficiently to cause any considerable hammering, which would tend to do damage in other ways.

In the case of hot bearings it is often advantageous to add a little powdered blacklead or sulphur to the oil, to assist in carrying off the heat generated by the friction. The blacklead or sulphur should be sifted through bunting, to cause it to be quite free from lumps and grit. If, after the preceding, the bearing still heats, the engine must be stopped and the bearing properly examined, lubricators cleared, and bearing refitted as necessary before attempting to again run at a high speed.

(*b*) When a bearing is *suddenly* found to be *hot*. The engine should be slowed down, and, if necessary, stopped, and the bearing be practically cooled with oil before the application of any water, as the immediate use of water may fracture the bearing through causing sudden local contractions. Caution is always necessary in the application of cold water.

Special precautions are required with all bearings when the engines are working with increased speed in the astern direction, as the normal conditions as to pressure are reversed. This specially applies to the thrust and crosshead guide bearings, where entirely different surfaces are brought into contact, and particularly to the latter, where the area of the astern working surface is generally considerably less than on the ahead side.

The bearings which require most careful watching are those in which one of the rubbing surfaces will soon become plastic when overheated, such as ordinary white metal, which melts at about 400° Fahr. The bearings so fitted are: crank-heads, main bearings, and plummer blocks almost invariably; and gudgeons or top-end bearings, eccentric straps, crosshead guides, and thrust bearings generally. The use of white metal is also being extended, with advantageous results. When fitted it is the practice at high speeds to assist the oil lubrication with a little water, which forms a lather. This lather is not only an efficient lubricant, but as it will neither form nor remain on a hot bearing it is a good visible guide to the working condition.

Use of water service.—Water should, however, be used as little as possible, as if not applied with care it corrodes and destroys the bearing surfaces. If in any case water is used on the bearings during the working of the engines, the supply should be discontinued some time before stopping, and oil only used instead, so that the journals may become coated with oil and preserved as far as possible from

rusting after the engines are stopped; also, the caps and brasses should be removed at the earliest opportunity for examination, and if used on the connecting-rod ends or main bearings the bolts should be drawn, cleaned, and coated with a preservative—either tallow or mineral oil and blacklead—before being replaced.

Stern tube bearing.—The condition of this bearing should be ascertained by feeling the bottom part of the gland, and testing the temperature of the water which runs through the cock fitted at the bulkhead. If warm, the circulation of water should be increased by slacking back the gland and opening the water service cock on the bulkhead as necessary.

CHAPTER XXXI.

ENGINES DONE WITH—EXAMINATIONS, ADJUSTMENTS, AND GENERAL INFORMATION.

Engines done with.—When the ship arrives in harbour and the engines are finally done with, the regulating and stop-valves should be closed, the worsteds taken out from the lubricators, cylinder and jacket drains opened, the engines *wiped down while warm* to clean off the grease, &c., adhering to them, and the bilges should be pumped out by one of the auxiliary engines. All sea connections should be closed except those necessary for any auxiliary machinery which may be kept in use, and the condenser casings and pipes drained to prevent corrosion. If the engines are not likely to be used for some time, all the water should be drained out of the condensers, hot wells, and feed-tanks, which should then be kept dry. The covers and doors should be removed as necessary, and need not be rejointed until it is necessary to again prepare for steaming.

If the ship has been for a considerable period under steam, advantage should be taken of the interval to make a thorough examination and adjustment of the working parts and to remedy all defects. The manhole doors on cylinder covers, or with small cylinders the covers themselves, should be removed, and the junk-rings of the pistons taken off for examination of the springs, &c. The pistons and the insides of the cylinders should be cleaned and oiled, and the junk-rings replaced. The manhole door should not be rejointed, so that the internal parts may be kept clean and free from rust until again required for steaming. The slide casing doors should be removed, slide-valves examined, and taken out if necessary, the surfaces cleaned and oiled, defects, if any, made good, packing rings adjusted or refitted, and covers rejointed. The packings of all the glands that have not been renewed for some time, or which show signs of leakage, especially those on the cylinders and condensers, should be examined, and renewed or refitted where necessary. The condenser doors should be taken off for examination of the packings at the ends of the tubes.

All the bearing surfaces, working parts, and fastenings should be examined and adjusted. The bolts of the bearings should be drawn back and thoroughly cleaned and coated with preservative as previously described The coupling bolts of the screw shafting should be examined, and some of them drawn back to insure that they are in good order.

The boilers and mountings, as practicable, should be examined, and defects made good, and all auxiliary machinery should be examined, adjusted, and repaired if required.

have been completed and defects made good, all main and auxiliary engines and gear should be kept clean and oiled, and partly turned round every day by the hand turning gear. The slide-valves of each of the engines should be worked daily by the starting gear, and levers and other working parts moved occasionally to prevent them from sticking. The sea and bilge cocks or valves of the pumps should be opened and closed daily, except those leading to the condenser, and the watertight doors, sluice-valves, Kingston, flooding, and other sea-cocks and valves, and the cocks or valves of the fire-main, should be worked regularly every week to insure their being kept in proper working order.

Examinations after being at rest a considerable time.—The 'Steam Manual' governing procedure in the Royal Navy requires that when a vessel has been lying in reserve the undermentioned parts of the machinery, with the gear attached to them, should be examined and, if movable, worked, to ascertain that they are in good order before steam is allowed to be raised, the result of the examinations being certified in writing by the officers who make them :—(1) Main and auxiliary stop-valves; (2) safety-valves; (3) brine-valves, pipes, and Kingston valves; (4) steam and water gauge cocks and pipes; (5) feed-cocks, valves, and pipes, drain-cocks and pipes to all boxes, and all other mountings; (6) separators, steam pipes, expansion and other joints; (7) stop, regulating, reducing, and slide-valves; (8) auxiliary starting valves, hand and steam starting gear, and all relief cocks and escape valves; (9) sea suction and discharge valves, and other valves and cocks in connection with the surface condensers, main-engine pumps, and auxiliary engines; (10) the shaft couplings, nuts, cotters, keys and pins connecting the working parts, and all other important fastenings of the machinery should be sounded and overhauled; (11) pistons, and especially fastenings of junk-rings; and (12) piston-rod nuts and guards, and any attachments to the pistons.

Adjustment of main bearings, crank-pins, and gudgeons.—This is a most important matter, and one that should always have the personal attention of the engineer in charge of the machinery. It is very necessary that the connecting rod and crank-shaft bearings should be kept properly adjusted, and that the engines should not be allowed to work with the journals slack. This would in the latter case leave the shaft improperly supported, while in the former case the hammering of the pistons on the crank-pins at each stroke causes a serious increase of bending strains on the crank-shaft which, from the nature of the case, it is difficult to reduce to exact calculation. Many broken shafts have been ascribed to this cause.

The main bearing and connecting-rod brasses should be screwed up tightly on to the stops or liners and not left loose. The corners of large nuts—i.e. for main bearings, crank-pins, crosshead or gudgeon bearings, &c.—should be numbered, and one or two well-defined lines marked on the caps, so that the positions of the various corners can be measured in relation to these lines and recorded. Knowing the pitch of the thread of bolts, the distance the corner of each nut requires to be moved through for an adjustment of $\frac{1}{200}$ of an inch or any other distance can be easily calculated. A record should be kept for all large bearings, which

will obviate repeated markings by centre-punch, chisel, &c., which is confusing and leads to error.

It is impossible to lay down a hard and fast rule for the amount of slackness from 'hard home' at which large revolving or oscillating bearing surfaces should be worked. This must always be a matter of experience, as all adjustments when cold are subject to alteration when under working conditions, due to various causes.

To adjust the amount of slackness in large bearings the use of lead wire is very common, a small piece of such wire $\frac{1}{16}$ inch diameter being inserted along the axis between the cap and the shaft or pin, and the nuts screwed up evenly till the thickness of the wire corresponds to the desired amount of play measured on a wire gauge. The position of the nuts should now be carefully observed, and after removing the wire the distance pieces and thin packing strips should be so fitted that when the nuts are screwed up tightly against the packing pieces the corners of the nuts are in the observed position. All white metal lined bearings should be worked with as little play as possible. When the engines are new a greater play is necessary than subsequently, when the bearings, after having worked some time, can be gradually adjusted for a smaller amount of play.

After adjusting crank-head and gudgeon bearings, they should be tested by moving them along the pins by means of a crowbar in several positions, to see that they are free, which of course they should be. Gudgeon brasses should be tested with the connecting-rod at its greatest angle with the line of centres, as these tend to wear oval, and the hardest parts must be quite free.

When time is important and there is a heavy knock which requires removal from a bearing, an old rule, expressed by 'halving the knock,' may be followed. This consists in removal of liners, tightening nuts hard home, noting the distance they have moved through, and then adjusting them slacked back to half this distance.

With horizontal engines great care is necessary before adjusting the bearings to insure that the shaft or pin is bearing hard against the bottom brass, otherwise there may be much more play than intended and shown by the lead wire. For this purpose it is necessary to turn the engines by the turning gear in such direction that the pins or shaft will be pressed against the back brass prior to the adjustment being carried out. With vertical engines the weight of the various parts tends to effect this object, but the engines should also be turned in this case.

Adjustment of other working parts.—Especial care is necessary with the link motion, as most of the wear takes place in the ahead position, so that only a portion of the wear can be taken up, and unless the links are trued up for all positions of the block there must always be a slackness in the ahead position.

Adjustment of eccentrics may best be effected by testing their freedom with the ends disconnected from the rods or links.

The crosshead slippers are generally adjusted by moving the crosshead as near as possible to the cylinder, displacing the piston-rod packing, and lining up till the rod is equidistant from the hole in the cylinder for the packing. In the fore-and-aft direction, however, the rod will, when cold, often be found not to be quite equidistant, owing

adjustment should also be tested when hot by turning the engine in such a direction that the slipper which is being adjusted is pressed home on the guide, and noting if the piston-rod is true for each point of the stroke. For this purpose it will be obvious on consideration that if the ahead slipper is being adjusted the turning gear must be worked so as to move the engine in the astern direction, and *vice-versâ*. Marks should be made on the crosshead when the engines are new and corresponding marks on the guide, with gauges fitted between them, which can be applied at any time, and which will also indicate the amount of lining up required.

The outsides of the crank and propeller shaft journals should always be marked when new, as a guide to indicate when the shafting is working forward from the wear of the thrust collars. From these marks the proper adjustment of the thrust-block is made in the fore-and-aft direction.

All bearings that have been adjusted should be carefully watched for a time after being under way.

Relief rings for flat slide-valves.—It is necessary that these rings should be properly adjusted, to insure steamtightness without too great a pressure on the working faces. If they be slack, the equilibrium of the valve is destroyed, which increases the stress on the eccentrics and gear, and a loss of efficiency will ensue from the passage of steam to the condenser or receiver from the back of the valve, without performing work in the cylinder. If too tight, the friction is increased, and unnecessary strains are brought on the eccentrics and slide-gear.

Pistons.—Great waste of steam will ensue if the pistons leak and allow steam to pass from the steam to the exhaust side, and thus, in the case of the low-pressure cylinder, to the condenser, without the performance of any work. The metallic packing ring should, therefore, always be kept pressed against the working surface of the cylinder. Advantage should frequently be taken of opportunities for taking off the junk-ring for inspection of the piston springs. In the case of horizontal engines, the pistons have a tendency to wear down and allow the steam to pass over them to the condenser. They and the guide surfaces should be kept lined up to the central position, and the metallic packing rings should be turned some distance round as they become worn. When horizontal pistons are fitted with back supporting rods or trunks, the guides should be lined up and adjusted as required, to prevent, as far as possible, the weight of the piston from resting on the bottom of the cylinder.

The condition of the pistons and valves should be tested for tightness, as the indicator diagram is of little or no service as regards this important defect, for leakage of steam, unless under exceptional circumstances, has so little effect on an indicator diagram that its detection by this means can seldom be effected.

The pistons and valves can, however, be tested when the engine is at rest by thoroughly warming it and then fixing the piston in some position and admitting steam to one side by the starting or pass-valve, and noticing the amount of steam passing to the other side by means of the indicator cock, or lifting the escape valve or drain cock. The

tightness of the valves can also be tested in this manner. Steam should be kept in the steam jackets while this is being done. Periodical testing in this manner and comparison with the original condition will often locate many defects.

Boiler tests.—To promote safety in the working of the boilers, and to serve as a guide for the reduction of the load on the safety valves when it may become necessary, it is desirable that the boilers should be tested by water pressure at regular intervals. To carry out this test, the boilers are first filled, care being taken that they are practically free from air, and the pressure is then produced by pumping additional water into them, by means of one of the steam pumps if fires are alight in other boilers, and, if not, by a hand pump or by one of the pumps specially supplied with fittings for this purpose. During the application of the water pressure, the boiler should be carefully examined and gauges used to detect any change of form in the furnaces and combustion chambers. The thickness of the plates should also be periodically ascertained by drilling small holes through them. The test-holes are afterwards tapped and filled with screw rivets.

The Admiralty rule is that, provided during the examination no indication of weakness is observed, the water pressure test should be double the working steam pressure in all boats' boilers, and in large water-tank boilers where the steam pressure is 90 lbs. per square inch and under; if above 90 lbs. and not over 155 lbs. steam pressure, 90 lbs. above the working pressure; and with water-tube boilers to $1\frac{1}{2}$ times the working pressure. If any sign of probable permanent deformation be detected, the test should be stopped, and the working pressure is, if originally not more than 90 lbs. on the square inch, then limited to two-fifths of the pressure arrived at before such indications were seen; and if the working pressure be originally above 90 lbs. and not above 155 lbs. per square inch, the reduced working pressure is not to be more than 90 lbs. less than, or, in the case of water-tube boilers, two-thirds of the test pressure arrived at before such indications were seen.

This reduced pressure is used until the defect, if local, can be made good and the proper test pressure applied. If the drill test should show unusual thinness in any part, the water pressure should be very carefully applied, to prevent injury being caused from over-pressure. In all cases it should be carefully noted whether there is permanent set. The Admiralty practice as regards amount of test pressure is followed generally by other foreign navies. The Board of Trade and Lloyd's, however, require a test of double the working pressure.

Ventilation of coal bunkers. Precautions to prevent accidents.—Considerable care is necessary to prevent accident from explosion of gas in coal bunkers. The coal-shoots should always be kept quite clear of coal to permit the gas to escape through the grated covers on the deck. Ventilating pipes are usually carried from the upper parts of the bunkers to the funnel casings, to allow the impure air and gases as they form, to pass away freely to the atmosphere. Inlet pipes are also fitted to admit fresh air, and these are led to the upper parts of the bunkers, as far as possible from the outlet orifices, so that the ventilation may be from the surface, and not through the body of the coal. The ventilating pipes from the several bunkers should be independent of each other, and always kept quite clear and open for ventilation,

special cases, where coal bunkers are not provided with permanent ventilating arrangements, comprising a separate inlet and outlet, care must be taken that the bunker lids are taken off frequently to keep the bunkers well ventilated. This should be done at least four times a week for not less than three hours at any one time.

The ventilation of bunkers, especially those without permanent ventilating fittings, should be tested by using safety lamps before sending men to work in them, and whenever the coal bunker lids are removed, lights should not be brought near the openings until the accumulated gas has been allowed to escape. Special precautions should be taken in this respect for a few days after coaling. Small tubes with screwed deck-plates are fitted in the bunkers at regular intervals, about ten feet apart, to enable the temperature to be ascertained. This should be done frequently, and if the temperature is rising means should be adopted for increasing the ventilation and getting rid of the gas.

Wet coal to be avoided.—Moisture sometimes causes a rapid generation of heat and gas, especially when the coal contains a considerable quantity of pyrites. Wet coal should not be shipped, and the coal should be kept as dry as possible after it is in the bunkers. Ships should not be coaled on rainy days if it can be avoided, and bunker lids should be replaced as soon as possible after coaling to prevent water passing into the bunkers when the decks are being washed.

Deterioration of Welsh coal.—This arises from two principal causes. The first is the oxidation of the organic constituents of the coal when exposed to air. This is known as the 'weathering of the coal,' and is greatly intensified by increase of temperature. The second cause is the very friable nature of Welsh coal, which causes it to easily break up when handled.

CHAPTER XXXII.

MATERIALS USED IN CONSTRUCTION.

THE most extensively used material in the construction of engines and boilers is iron, using the term in its inclusive sense to comprise cast-iron, wrought-iron, and steel; which, though differing so greatly in qualities, are but different forms of the same material.

Iron is very rarely found in the metallic state, but is generally combined with oxygen and carbonic acid, and mixed to a greater or less extent with clay and earthy matters. In this condition it is called *iron ore*, of which there are many varieties.

British iron is made from the ores known as red hæmatite, clay ironstone, and black-band, but principally from the two latter.

Cast-iron.—Cast-iron is obtained from the iron ore by the process known as 'smelting.' The substances employed in smelting are: (1) the ore itself; (2) the fuel, which produces heat by its combustion and supplies carbon; (3) the air, which supplies oxygen for the combustion of the fuel, and for combination with the carbon in the ore; and (4) a flux, generally lime, which promotes fusion of the ore and combines with the earthy portion, forming a slag.

The iron, after it is reduced from the ore, is drawn off from the blast-furnaces, run into a series of shallow gutters or grooves, and broken into short pieces, about 2 or 3 feet long. It contains in its composition a proportion of carbon, from 3 to 5 per cent., and is known as *pig-iron* or *cast-iron*. Only a part of this carbon is actually in chemical combination with the iron—say, from 1 to about $2\frac{1}{2}$ per cent.—the remainder being diffused throughout the mass in the form of graphite or plumbago.

From its low first cost, its strength, and the facility with which it can be cast into any form, *cast-iron* is extensively used in all engineering work. In the marine steam-engine, the cylinders and covers, slide casings and valves, framing, plummer blocks, and many other parts, especially those of intricate form, are generally made of cast-iron, and often the pistons and condensers. In land and mercantile marine engines the stop- and safety-valve boxes are also usually made of cast-iron, and in land engines the steam-pipes also.

The properties of different brands of cast-iron vary very widely according to the quality of the ore from which they are produced, and to the proportion of carbon actually combined with the iron. The iron that contains the greater quantity of carbon in combination is called *white cast-iron*, from the appearance of the fracture. It is very hard and brittle, and unsuitable by itself for foundry purposes. It is known in the market as No. 8 pig.

At the other end of the scale is *grey cast-iron* (No. 1 pig), in which

small particles of blacklead or graphite, which give to the fracture a greyish colour. This is much softer and tougher than the white iron, and is generally used in making castings, being mixed with some of the whiter varieties, or with good scrap cast-iron, to give it sufficient strength and hardness for various purposes. Cast-iron with properties intermediate between those of grey and white iron is often called *mottled cast-iron*, and its nearness in composition to grey or white pig is indicated by numbers intermediate between 1 and 8.

Cast-iron is improved in strength and closeness of texture by re-melting in the foundry cupola, the quantity of uncombined carbon being thereby reduced. The grey pig-iron as received from the blast-furnaces is not altogether suitable, alone, for engine castings, especially in cases where hard and smooth working surfaces are required. In such cases it is desirable to mix with it a proportion of good scrap cast-iron from old engine castings, to increase both the strength and uniformity of the casting. The proportion of scrap varies from 30 to 70 per cent. according to the degree of hardness required.

Unequal contraction in cooling.—The great objection to the use of cast-iron, especially for parts that have to sustain severe and intermittent stresses, is the uncertainty that exists as to its actual strength in any particular instance, in consequence of the unequal and unknown stresses brought on the material during the process of cooling in the mould. The initial stress is sometimes so great that the casting is found fractured on being taken out of the mould, before it has been subjected to the action of any external force, but the casting must be weakened in any case. The amount of contraction varies with the size and thickness of the casting and with the quality of iron used. In thin castings the contraction is about $\frac{1}{16}$-inch per foot, whilst in thick castings it is as much as $\frac{1}{8}$-inch per foot.

To prevent unequal contraction as far as possible, sudden variations in the thickness of the several parts should be guarded against, and sharp corners should be avoided. Suitable arrangements should also be made to cause the rate of cooling of the different parts of the casting to be as nearly uniform as possible. Unless proper precautions are taken this unequal contraction will often cause distortion of form in castings of irregular shape, and it is therefore an advantage to make the castings as symmetrical in form and uniform in thickness as is consistent with the purposes for which they are required.

Cast-iron is also liable to have its strength reduced by the existence of *blow-holes* or gas-bubbles underneath the surface, which cannot generally be discovered by any ordinary inspection or test, if they should be at any depth below the surface. If near the skin of the iron, they may be discovered by tapping the casting with a hammer.

In consequence of these defects, cast-iron is an unreliable material for structures of irregular form that have to sustain intermittent heavy loads, and a large margin of strength should be allowed when it is employed. It has, however, the advantages of cheapness and stiffness, and at present it is the only material that can be used for many parts of the machinery. If it can be avoided it should not be used in parts that have to withstand unequal and irregular temperatures; as it is an

unreliable material for parts that are exposed to unequal temperatures or subject to blows.

Several serious accidents have occurred from the bursting of cast-iron stop-valve boxes when steam has been admitted to them without previously taking the precaution of draining out the cold water that had accumulated in the valve-boxes and steam-pipes. For this reason gunmetal is generally used in the Navy for stop- and safety-valve boxes of marine boilers, but they are sometimes of cast-steel. In the mercantile marine they are commonly of cast-iron.

The strength of cast-iron under the action of a crushing load is much greater than when exposed to tension, its resistance to crushing being from 80,000 to 110,000 lbs. per square inch, whilst its average tenacity is only from 16,000 to 18,000 lbs. per square inch. It is, therefore, more suitable for parts that are exposed to compression than for those that have to sustain stretching or tension.

For the cast-iron used by the Admiralty the minimum tensile strength is 9 tons per sq. inch, and the minimum transverse breaking load for a bar 1 inch square loaded at the middle between supports 1 foot apart at least 2,000 lbs.

The soundness or compactness of a casting is promoted by casting it under pressure. Consequently cylinders, pipes, &c., should be cast in a vertical position, with a *head* or additional column of metal above, whose weight serves to compress the mass of metal in the mould below. The dross and gas-bubbles ascend into the head, which is cut off when the casting is cool.

Malleable cast-iron.—By imbedding an iron casting in oxide of iron, or powdered red hæmatite, which consists almost entirely of peroxide of iron, and keeping it at a high temperature for a sufficient time, which will vary with the size of the casting, a portion of the carbon contained in the iron will unite with the oxygen in the oxide, and the casting will be converted, to a greater or less extent, into a material resembling mild steel or wrought-iron. This material is much cheaper than wrought-iron or steel; but the process is only applicable to comparatively small articles of fairly uniform thickness.

Malleable cast-iron is not generally used for important parts of machinery, but the junction-boxes for the Belleville boiler tubes are made of malleable cast-iron, and the material gives satisfaction in this situation, although exposed to high pressures and temperatures.

Wrought-iron.—Wrought-iron, in its pure state, is simply metallic iron, without admixture or combination with any other element, and is produced from cast-iron by the processes of refining and puddling, or by the use of a Bessemer converter.

Wrought-iron was at one time universally employed for the construction of the boilers, shafting, piston and connecting-rods, and nearly all the moving parts of the engines, especially those that are subject to severe and varying strains. It is tough and strong, and has a fibrous structure, which renders its resistance to tension much greater than its resistance to compression. It is malleable and ductile, and though it can only be fused with difficulty and at a very high temperature, it possesses the property of *welding*, when raised to a white heat (say 1,500° to 1,600° Fahr.), which enables two pieces of iron to be firmly united or welded together by hammering.

The fibrous structure which the best wrought-iron shows is produced by the processes of manufacture, but vibration tends to reduce it to the crystalline condition, and this probably accounts for numerous fractures of parts subject to vibration and intermittent strains, the fractures in most cases showing crystalline or granular surfaces. General experience shows that if wrought-iron shafts, such as railway axles, marine-engine shafting, &c., which are exposed to intermittent stresses, be designed with the ordinary factor of safety used for machinery, they are liable to fracture after running for a certain period, and in such cases the appearance at the fractured part is crystalline. These parts are, therefore, generally made larger than at first sight appears necessary, so as to give increased stiffness and reduce the intensity of the torsional stresses on the particles.

The tensile strength of good iron forgings made from scrap-iron is about 22 tons per square inch with the grain and 19 tons across the grain.

Case-hardening.—The outer skin in many wrought-iron pieces of machinery is made hard, to resist friction, by the process of case-hardening, which consists in imbedding the article in some carbonaceous substance, and raising it to a red heat, by which means the outer layers acquire sufficient carbon to convert them into hard steel. One of the most convenient is finely powdered yellow prussiate of potash, with which the iron is sprinkled, heated to redness without access of air, and afterwards cooled in water. The depth of the hardening will depend on the time occupied in the process. In marine engines, the pins of the link motion and many other similar parts and small nuts are usually made of iron, case-hardened, or of steel. The gudgeon pins of wrought-iron or mild steel are also generally so treated when they work on a gunmetal surface.

Steel.—The term *steel* is applied to all compounds of iron and carbon in which the proportion of combined carbon does not exceed 1·5 per cent. The properties of the materials thus included under a common name vary, however, very greatly, according to the amount of carbon they contain. When the percentage of carbon is below 0·5, the material is called *mild steel*, and possesses few of the qualities popularly attached to steel, and in point of fact very closely resembles the best wrought-iron. The hardness and tenacity of steel, and its capability of fusion, increase as the percentage of carbon becomes greater.

Steel is produced in general either by the addition of carbon to wrought-iron, or by the abstraction of carbon from cast-iron. The former method, although more complex and expensive, is preferred for making the higher classes of steel required for tools, &c., as wrought-iron can be obtained in a greater state of purity than cast-iron. The second method is employed for making large quantities of steel rapidly and cheaply, such as that required for ordinary plates, bars, rails, &c.

Steel is distinguished from wrought-iron by its capability of being cast into a malleable ingot, so that uniformity of structure may be insured; and, above a certain percentage of carbon, also by its possessing the property of *tempering*, which enables it to be hardened by sudden cooling, a property valuable in the manufacture of cutting

tools; or softened by gradual cooling from a high temperature. The mild steel used for engine forgings and boiler plates, which only contains from 0·15 to 0·30 per cent. of carbon, is destitute of hardening qualities.

Bessemer steel.—In the Bessemer process for making steel, molten pig-iron is poured into a vessel called a *converter*, through which a stream of air is blown by a strong blast. The oxygen of the air first removes the silicon and manganese, and then unites with the carbon and carries it away. After all the carbon has been removed, the proper proportion of carbon required to make the steel is introduced by the addition of a quantity of special molten pig-iron (*Spiegel-Eisen*). The steel thus made is poured into ingots, and afterwards hammered, rolled, and worked as required. The process has to be a rapid one, however, and it is difficult to judge as to the correct time to stop the action of the blast. Irregularities are often found in the finished product.

Acid and basic Bessemer processes.—There are two Bessemer processes, depending on whether the converter has an acid lining of 'ganister' or whether it has a basic lining of 'dolomite.' The resulting steel is termed either 'acid' or 'basic' Bessemer steel. The difference consists principally in the action on any phosphorus contained in the molten pig-iron. With the basic lining the phosphorus is eliminated by the presence of lime in the converter.

With the acid lining, however, the lime cannot be added, as they combine with each other, so that the phosphorus remains. The pig-iron used with this process, therefore, must not contain more than the amount of phosphorus allowable in the steel; while with the basic process large quantities of phosphorus are permissible in the pig-iron.

Siemens-Martin steel.—In the *open hearth* process of making steel the great heat produced by a Siemens regenerative furnace dissolves in a bath of molten pig-iron the ores of iron, either in a raw state or in a more or less reduced condition. The oxygen in the ore unites with the carbon of the molten pig-iron to form carbonic oxide, which passes off as gas. Usually, steel or wrought-iron scrap is added in addition to the iron ores. The principal advantage of this system of producing mild steel is that it is not dependent on a limited time for its results, as is the case in the Bessemer process. The heat of the furnace is such that the fluid bath of metal, after having been reduced to the lowest form of decarbonisation, may be maintained in that condition for any reasonable length of time, during which samples may be taken and tested, and such additions made to it as may be necessary to adjust it to the required quality, and uniformity in all the ingots produced may thus be insured. To further improve the quality of the material, a small quantity of *Spiegel-Eisen* and ferro-manganese is generally added. There are both acid and basic processes for Siemens-Martin steel, the differences being similar to those described above for Bessemer steel.

Properties of mild steel.—Mild steel can be worked well at a red heat, and can be bent cold into most ordinary forms. It, however, possesses the peculiar property of becoming brittle at a temperature between about 400° and 600° Fahr., which is technically known as a *blue heat*, from the colour of the fracture at that temperature. Care

material after it has fallen to this dangerous limit of temperature.

All plates or bars that have had much work done on them while hot should be subsequently annealed. The annealing, if possible, should be performed simultaneously over the whole of the plate or bar in question, and care should be taken to prevent the access of air to the furnace, or the impinging of the flame on the material during the process. After the material has been raised gradually to a red heat, it should be taken out of the furnace and allowed to cool slowly. Annealing is not necessary in cases where the whole of the plate has been heated and bent or flanged at one heat.

Steel made by the Siemens-Martin process is now extensively used for engine forgings and for boiler plates and tubes, instead of wrought-iron. To insure soundness the ingots from which the plate or forging is made should be cast with a large head. During the solidification the lower part of the ingot has to feed itself from the head, forming a funnel-shaped cavity usually known as 'the pipe,' which is essential for the ingot to be sound. The head must be long enough to allow the whole of this pipe to be cut away and the forging to be made from the sound metal below it.

The tensile strength of steel used in boiler work is between 27 and 30 tons per square inch, with an elongation in 8 inches of length of not less than 20 per cent. before fracture. For steel forgings the Admiralty permit the strength to be between 30 and 35 tons, with an elongation of 27 per cent. For crank and propeller shafting the limits are between 28 and 32 tons, with 30 per cent. elongation.

Whitworth's fluid compressed steel. — Sir Joseph Whitworth's system of producing sound steel ingots, free from blow-holes and suitable for the best engine forgings, consists in subjecting the metal while setting in the ingot mould to great hydraulic pressure, and by this method steel of very great uniformity and strength is produced. This has been extensively used for shafting, cylinder liners, and many other purposes. In the Whitworth system the whole of the forging is performed by suitable and powerful hydraulic presses, no hammering whatever being employed, but the ingot gradually squeezed to the required form. There can be little doubt that this system is superior to the use of steam hammers, and it is now the common practice of other leading steelmakers.

Steel castings.—In modern engines the frames, bedplates, pistons, and many other parts have been made of mild cast-steel, and this has enabled reductions of weight to be effected.

The greatest difficulty to be overcome in making mild steel castings is to prevent 'draw' in the metal from the contraction while cooling in the mould. The steel requires a temperature of about 4,000° Fahr. to melt it, as compared with 2,000° Fahr. for cast-iron, and the contraction of the steel casting is as much as $\frac{5}{16}$-inch per foot. Steel castings cannot be 'fed' as in the case of iron ones, and consequently the head must be of sufficient size and suitably arranged to allow of this feeding action taking place, and the milder the steel the greater this will be. This forms the 'pipe' in the head, which is generally an index of a sound casting, for if the casting in contracting does not feed from the head, it must feed from itself and become unsound. All

steel castings after being taken from the mould should, without first being allowed to cool, be reheated and annealed to insure molecular equilibrium and freedom from internal strains.

For ordinary steel castings the Admiralty require the strength to be between 28 and 35 tons per square inch, with elongation in 2 inches of not less than 15 per cent.

Nickel steel.—The tensile strength and limit of elasticity of mild steel is very considerably increased by adding to it a certain percentage of nickel. This percentage varies considerably according to the use the machine is intended for, but is generally from 3 to 5 per cent. With ordinary steel this increase of strength would be accompanied by a diminution of ductility, but with nickel steel this is not the case, and the stronger material is also very ductile. These properties appear to indicate its special suitability for many purposes, such as marine engine shafts, &c. It has already been used in many cases for such parts, and the combined material is gradually coming into more extended use, although experience with it is still limited.

After iron and steel, the materials most generally used in the construction of the machinery are copper, tin, zinc, lead, and their alloys.

Copper.—This metal is red in colour and very soft, malleable, and ductile when cold. It cannot be welded, and does not make good castings. It can, however, be readily worked cold, and it is consequently used principally for making steam and other pipes which require to be bent cold. At high pressures and temperatures large copper steam pipes have been found liable to split, and many accidents have occurred recently owing to this. In such pipes in the Navy steel is now always used, or the copper steam pipes are sometimes lapped round with copper wire secured at the ends.

For most purposes it is too soft and weak to be used by itself, but it is the principal element used in forming the various alloys included under the terms 'gunmetal' and 'brass,' which are so extensively used in various parts of the machinery.

Although the term 'brass' is often applied indiscriminately to all alloys of copper with tin or zinc, its use, strictly speaking, should be confined to alloys of copper and zinc only, those made with copper and tin being known as gunmetal or bronze. A little zinc is usually added to the gunmetal alloys to facilitate casting.

Gunmetal or bronze.—This alloy is considerably harder than copper, and offers much greater resistance to crushing, which makes it suitable for many parts of machinery. It is easily fusible, and forms good, sound, and strong castings. It is therefore extensively used in the marine steam-engine for making cocks and valves of all descriptions, condenser and other pumps and fittings. Gunmetal is much used for bearing surfaces in machinery, as it is sufficiently hard and durable to prevent excessive wear, but less so than iron or steel, so that the bearing will wear instead of the journal. The alloys of copper and tin increase in hardness and brittleness as the percentage of tin is increased. The ordinary gunmetal used in machinery is much tougher than cast-iron, and is, therefore, more suitable for parts that are subject to shocks. In consequence of its resistance to corrosion, it is very suitable for pumps, valve-boxes, propellers, and other

fittings in preference to cast-iron, though its first cost is much greater.

The proportions of copper, tin, and zinc in the gunmetal ordinarily used vary to some extent with the nature of the article produced. In the Navy the proportions generally are copper 88, tin 10, zinc 2. For engine bearings the proportion of zinc may be increased and copper decreased, the following being an analysis suitable for this purpose: copper 85, tin 10, and zinc 5. The tensile strength of good gunmetal, such as that used for bolts, &c., may be taken at about 10 to 14 tons per square inch. The Admiralty specify their gunmetal to be of a tensile strength of at least 14 tons per square inch with an elongation of 7½ per cent. on a length of 2 inches.

Brass.—Ordinary brass which is used for cheap castings, where strength is not important, is an alloy composed of about two parts of copper to one of zinc, and is yellower in colour and much softer than gunmetal. It is not suitable for parts exposed to compression, and is not used for making important engine castings.

Brass tubes.—If the proportions of copper and zinc in the alloy be suitably arranged, the brass produced will be malleable, and may be rolled into sheets or drawn into tubes or wire. The condenser-tubes and internal steam and feed-pipes in the boilers are generally made of this material. The proportion of copper in the condenser tubes is generally 70 per cent., with the addition of 1 per cent. of tin to prevent corrosion from sea-water, the remainder being zinc.

Muntz-metal.—This is composed of about 60 parts of copper to 40 parts of zinc, with frequently 1 per cent. of lead to assist malleability. This material can be rolled hot into bars, plates, and sheets, and has been largely used for making rods, bolts, &c., as it possesses considerable tenacity. It has, however, been found that if Muntz-metal bolts are in contact with copper or gunmetal in sea-water, galvanic action ensues, which speedily decomposes the Muntz-metal, the zinc in the compound being destroyed. It is equal in tenacity to that of good wrought-iron.

Naval brass.—The decay of Muntz-metal referred to in the previous paragraph appears to be very largely reduced or practically prevented by the addition of a little tin to the metal. An alloy composed of 62 parts of copper, 37 parts of zinc, and 1 part of tin, which for distinction is known by the name of *naval brass*, is now used instead of Muntz-metal for all such fittings. The addition of the tin does not prevent the naval brass being forged with quite as much facility as Muntz-metal. Naval brass bars or sheets have a tensile strength of at least 26 tons per square inch.

There are various descriptions of special bronzes used which have a strength considerably above that of ordinary gunmetal.

Manganese bronze consists of naval brass with the addition of a small proportion of ferro-manganese. Its strength when rolled is about 28 to 30 tons per square inch, but can be increased to 35 tons. This is a favourite metal for propeller blades.

Phosphor bronze, consisting of copper and tin with a little phosphorus added, is also considerably used.

Aluminium copper and bronze.—Aluminium when added to copper sheets and brass improves their qualities in a marked degree, especially

as regards reduction of corrosion. Various alloys, termed aluminium copper and bronze, are now being used in the Navy, and they seem destined to be still more useful to engineers in the future. Aluminium copper pipes are composed of about 2 per cent., or slightly more, of aluminium, the remainder being pure copper. This percentage of aluminium should be present in the pipes after manufacture. It then has a tensile strength of at least 15 tons per square inch.

White metal for bearings.—A considerable number of different compositions for this purpose have been tried, but recent experience indicates that for large bearings carrying a heavy load, the following composition is satisfactory, *viz.* : tin 87 per cent., copper 5 per cent., and antimony 8 per cent. Slight variations from these exact proportions may be made by using from 2 to 7 per cent. of copper, the tin being adjusted accordingly.

Zinc.—Besides its uses in various alloys (brass, Muntz-metal, &c.), connected with the machinery, slabs of zinc are used for the protection of boilers. Rolled zinc is preferable to cast zinc for this purpose. Zinc is also used for coating iron that is exposed to the action of water or moisture, thus preventing its corrosion and decay. The zincing is performed by immersing the iron article, after it has been thoroughly cleaned by fire or acids, in a molten bath of zinc, by which means it becomes coated with a layer of zinc, which increases in thickness according to the length of time that the article is kept in the bath. This is termed zincing or galvanising by the 'hot' process.

The weldless steel boiler tubes now being used in new water-tube boilers for the British Navy are generally coated externally with a thin coating of zinc electrically deposited. This process shows up some defects in tubes which are difficult to discover otherwise, and the coating of zinc aids in preserving the tubes during the construction of the boilers. In the British Navy, only the outside surface is so coated, and this is effected in the following manner :—The tubes are first cleaned bright by being pickled in weak hydrochloric acid till all scale formed during manufacture is removed, after which, and immediately before zincing, the tubes are immersed for about twenty minutes in a weak sulphuric acid bath, the ends of the tubes being plugged during the latter process. On removal from the sulphuric acid the tubes are well brushed and thoroughly washed. This is known as the 'cold' process. The internal surfaces of some types of boiler tubes are also galvanised by the hot or dipping process to prevent or arrest corrosion. Economiser tubes of Belleville boilers are in the later examples treated in this manner.

Lead.—Lead is not used alone in any part of the marine engine, except occasionally in the form of weights for balancing certain moving parts, but it is used in combination with other soft metals for making the white-metal alloys for bearing surfaces.

CHAPTER XXXIII.

THEORETICAL INDICATOR DIAGRAMS OF STAGE EXPANSION ENGINES.

To determine the relative proportions of the cylinders of stage expansion engines, and the points of cut-off in each to produce the most uniform strains on the shafting, and enable the work due to the expansion to be most fully realised, theoretical indicator diagrams showing the action of the steam in the several cylinders are very useful. For simplicity, we will neglect the effects of the release before the end of the stroke, the compression, the resistance of the passages between the cylinders, &c., simplicity being of more importance than extreme accuracy, especially as allowance can easily be made for the compression, early release, &c., after the diagrams have been drawn, if the principles involved are clearly understood.

Let V represent the volume of the large cylinder;
v the volume of the small cylinder;
U the volume of the intermediate reservoir;
R = total rate of expansion;
Rate of expansion $= r$ in high-pressure cylinder, and ρ in low-pressure;
λ = ratio of cylinders; so that $R = r\lambda$;
ϕ = ratio of reservoir to h.p. cylinder; so that $U = \phi v$;
p_1 = initial absolute pressure in the high-pressure cylinder.

Since a volume $\frac{v}{r}$ of steam entering the high-pressure cylinder at pressure p_1, occupies finally a volume V at pressure $\frac{p_1}{R}$, we shall have $\frac{v}{r} = \frac{V}{R}$.

Compound engines with cranks at 0 deg. or 180 deg. apart, but without an intermediate reservoir.—In Fig. 386, O B represents the initial absolute pressure of the steam on its admission to the high-pressure cylinder, and B C is the line of pressure during admission. At C the steam is cut off and expands in the small cylinder to D, the end of the stroke, when the communication is opened to the large cylinder, and the steam exerts a forward pressure on the large piston and a back pressure on the small piston. This part of the action of the steam is represented by the two curves, D A and E F, the ordinates of D A representing the back pressures on the small piston and the corresponding ordinates of E F the forward pressures on the large piston. O P is $= V$, the volume of the large cylinder, and O N $= v$, the volume of the small cylinder. At the end of the stroke of the large piston the communication is opened to the condenser, and the pressure falls to P G, the constant condenser pressure. These diagrams may be combined as follows:—Draw any straight line, $a\ b\ c\ d$, parallel to P O Q, and intersecting the two diagrams, and lay off on it $c\ d = a\ b$,

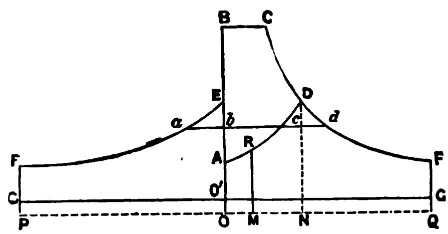


Fig. 386.

then $bd = bc + cd$ represents the total volume occupied by the steam when its absolute pressure is Ob, and 'd' is a point on the indicator diagram which would be formed if the steam had been expanded in the large cylinder only. By drawing a sufficient number of horizontal lines and laying off the proper distances on them, any number of points can be found, and the diagram can be reasoned about as if the whole of the action had taken place in one cylinder only. The pressure at any point in the forward stroke of the large piston, and back stroke of the small piston, is easily obtained. At any point M in the return stroke of the small piston, the total volume occupied by the steam is $(v - x) + x\frac{V}{v}$, where $x = \text{NM}$.

Therefore the pressure M R

$$= p_1\frac{v}{r} \div \left\{v - x + x\frac{V}{v}\right\} = p_1\frac{v}{r} \div \left\{v + x\left(\frac{R}{r} - 1\right)\right\} = \frac{p_1 v}{vr + x(R - r)}$$

Compound engines with cranks at 0 deg. or 180 deg. apart, but with an intermediate reservoir.—This is a case that seldom occurs in practice unless the reservoir be used for the purpose of reheating the steam on its passage from the high- to the low-pressure cylinders. It is, however, interesting to examine the effect of the reservoir on the diagram, because in any case the passages between the cylinders form a sort of reservoir—in some cases not an inconsiderable one.

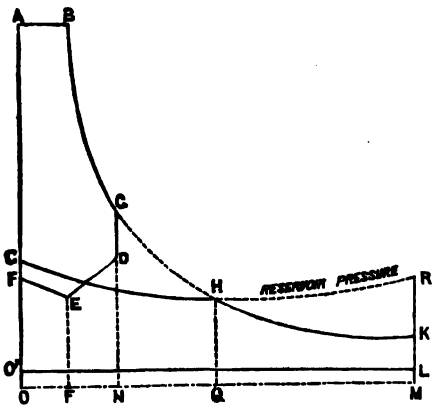


FIG. 387.

Let p_r = pressure in the reservoir immediately before the high-pressure cylinder exhausts into it. O A, Fig. 387 = p_1 = initial absolute pressure of the steam in the high-pressure cylinder. At B the steam is cut off and expands to C, the end of the stroke of the high-pressure cylinder. At this point the communication is opened to the reservoir, and a volume, v, of steam at pressure $\frac{p_1}{r}$ is admitted to the reservoir; consequently the pressure N D will be $= \dfrac{p_r \text{U} + \frac{p_1 v}{r}}{v + \text{U}}$. This is, of course, equal to the initial pressure O G in the low-pressure cylinder. The steam now acts on the low-pressure piston until $\frac{1}{\rho}$th of the stroke of the low-pressure piston has been performed, when the admission to the large cylinder is cut off. At this point the steam occupies the volume $\left(1 - \frac{1}{\rho}\right)v + \text{U} + \frac{\text{V}}{\rho}$, and its pressure is therefore

$$= \frac{p_r \text{U} + \frac{p_1}{r}v}{\left(1 - \frac{1}{\rho}\right)v + \text{U} + \frac{\text{V}}{\rho}} = \frac{p_r \phi + \frac{p_1}{r}}{\left(1 - \frac{1}{\rho}\right) + \phi + \frac{\lambda}{\rho}}.$$

This part of the action of the steam is represented by the curve G H in the low-pressure diagram, and D E in the high-pressure diagram. After

the steam is cut off, it expands in the cylinder to the final pressure $\frac{p_1}{R}$, while in the reservoir it is compressed to the pressure p_r, represented by $MR = OF$. It only remains to determine p_r in order that the diagrams may be completely drawn. We can easily find p_r from the fact that a volume $\frac{V}{\rho}$ of steam at pressure $\dfrac{p_r\phi + \frac{p_1}{r}}{\left(1 - \frac{1}{\rho}\right) + \phi + \frac{\lambda}{\rho}}$ occupies finally a volume V at pressure $\frac{p_1}{R}$.

$$\text{Therefore } \frac{p_r\phi + \frac{p_1}{r}}{\left(1 - \frac{1}{\rho}\right) + \phi + \frac{\lambda}{\rho}} \cdot \frac{V}{\rho} = \frac{p_1}{R} V$$

$$\text{or, } p_r\phi + \frac{p_1}{r} = \frac{\rho p_1}{R}\left\{\left(1 - \frac{1}{\rho}\right) + \phi + \frac{\lambda}{\rho}\right\}$$

$$\text{but } R = \lambda r$$

$\therefore$ by substitution and reduction we get $p_r = \frac{p_1}{R}\left\{\left(\frac{\rho - 1}{\phi}\right) + \rho\right\}$

The diagram can now be completely drawn. If the reservoir pressure p_r be not so great as the pressure of release in the high-pressure cylinder, $\frac{p_1}{r}$, there will be a fall of pressure on the admission to the reservoir, and the work due to expansion will be partly lost. If these pressures be equal we have:—

$$\frac{p_1}{r} = \frac{p_1}{R}\left\{\frac{(\rho - 1)}{\phi} + \rho\right\}$$

$$\text{or } \frac{R}{r} = \lambda = \frac{\rho - 1}{\phi} + \rho \text{; therefore } \rho = \frac{\phi\lambda + 1}{\phi + 1}.$$

From this equation in any given case ρ can be determined, so that there shall be no loss on admission to the reservoir. When $\phi = 0$, $\rho = 1$; this is the case previously discussed. Taking $\phi = 1$, that is, taking the volume of the reservoir equal to that of the high-pressure cylinder, we have $\rho = \frac{\lambda + 1}{2}$. In this case, if λ be greater than 3, ρ will be greater than 2, consequently arrangements should be fitted to cause the cut-off in the low-pressure cylinder to be before half-stroke if the work due to the expansion is to be fully realised.

The following table gives a few values of rates of expansion necessary in the low-pressure cylinders of compound engines of this type when there is no fall of pressure on the admission to the reservoir.

$\lambda =$		1	2	3	4	5	6
ρ for	$\phi = 1$	1	$\frac{3}{2}$	2	$\frac{5}{2}$	3	$\frac{7}{2}$
	$\phi = 2$	1	$\frac{5}{3}$	$\frac{7}{3}$	3	$\frac{11}{3}$	$\frac{13}{3}$
	$\phi = 3$	1	$\frac{7}{4}$	$\frac{10}{4}$	$\frac{13}{4}$	4	$\frac{19}{4}$

We now pass on to consider the type of compound engine most generally used, viz. engines with two cylinders, side by side, acting on cranks at right angles to each other, and having an intermediate reservoir. The cases in which the cut-off in the low-pressure cylinder is after half-stroke, and before half-stroke respectively, must be discussed separately.

Two cylinder compound engines with cranks at right angles to each other, having an intermediate reservoir, the cut-off in the low-pressure cylinder being after half-stroke.—It will be necessary in the first place to find an expression for the distance of the high-pressure piston from the end of its stroke, when the steam is cut off in the low-pressure cylinder.

Let O A (Fig. 388) be the position of the crank of the low-pressure cylinder

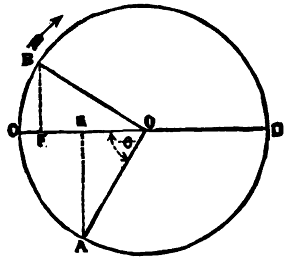


FIG. 388.

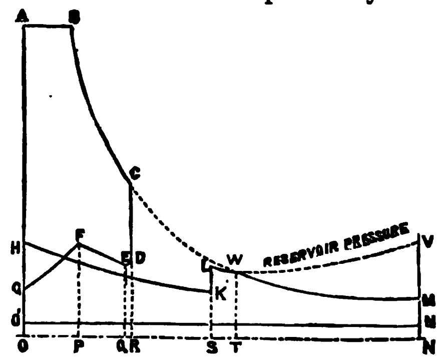


FIG. 389.

when the steam is cut off; O B the corresponding position of the crank of the high-pressure cylinder.

Then $\frac{\mathrm{D\,E}}{\mathrm{D\,C}} = \frac{1}{\rho}$ and $\frac{\mathrm{C\,F}}{\mathrm{D\,C}}$ = fraction of stroke performed by the high-pressure piston, when steam is cut off in the low-pressure cylinder $= \frac{1 - \sin\theta}{2}$

$$\frac{\mathrm{D\,E}}{\mathrm{D\,C}} = \frac{1}{\rho} = \frac{1 + \cos\theta}{2}$$

$$\text{from which, } \cos\theta = \frac{2 - \rho}{\rho} \text{ and } \sin\theta = \sqrt{1 - \cos^2\theta} = \frac{2}{\rho}\sqrt{\rho - 1}$$

$$\text{Therefore, } \frac{\mathrm{C\,F}}{\mathrm{D\,C}} = \frac{\rho - 2\sqrt{\rho - 1}}{2\rho}$$

In the following investigation we will denote this by m; consequently the fraction of the small cylinder that is occupied by the steam that acts on the low-pressure piston at the point of cut-off is $= (1 - m)$.

The following table gives some values of $(1 - m)$ for different values of ρ :—

$\rho =$	·5	·55	·6	·65	·7	·75	·8
$1 - m =$	1	·998	·990	·977	·958	·933	·9

In Fig. 389, O A represents the initial pressure of steam in the high-pressure cylinder. At B the steam is cut off and expands to C, the end of the stroke of the high-pressure piston, when the communication is opened to the reservoir and the pressure falls to R D. After this the steam expands in the reservoir and low-pressure cylinder until it is cut off in the latter. This part

pressure at the point of cut-off. From this point the steam is compressed behind the high-pressure piston until it has completed half its return stroke, when its pressure is represented by P F. At this point the admission to the low-pressure cylinder commences, and the steam expands in the low-pressure cylinder until the end of the return stroke of the high-pressure cylinder, when its pressure is O G.

The low-pressure diagram is easily deduced from this. The initial pressure O H is, of course, equal to the back pressure P F at the middle of the return stroke of the high-pressure piston. The steam expands in the low-pressure cylinder until half-stroke, when its pressure, S K, is obviously equal to O G. At this point the high-pressure cylinder, containing steam at the pressure R C, opens to the reservoir, and the pressure rises to S L, S L being equal to R D. From L the steam expands in the reservoir and low-pressure cylinder to W, the point of cut-off, T W being equal to Q E. From W the steam in the cylinder expands to the final pressure N M, while that in the reservoir is compressed to V, N V being equal to the initial pressure in the low-pressure cylinder. At M the communication to the condenser is opened, and the pressure falls to N N′,—the constant condenser pressure.

We will now give the algebraical expressions for the pressures at the different points, in order that the diagrams may be drawn in any given case. Since the total rate of expansion is R, the final pressure, N M, in the low-pressure cylinder is $= \frac{p_1}{R}$. The final pressure, R C, in the high-pressure cylinder is $= \frac{p_1}{r} = \frac{p_1 \lambda}{R}$.

The steam in the low-pressure cylinder is expanded ρ times: consequently at the point of cut-off, the pressure, T W, is $= \frac{p_1 \rho}{R}$. This is also the pressure, Q E, in the reservoir at the point of cut-off, and we have, therefore, steam at the pressure $\frac{p_1 \rho}{R}$ occupying a volume $U + v\,(1-m)$. This steam is compressed behind the high-pressure piston until the beginning of the next stroke of the low-pressure piston, when its volume has been reduced to $U + \frac{v}{2}$, and its pressure has been increased to

$$\frac{p_1 \rho}{R} \cdot \frac{U + v\,(1-m)}{U + \frac{1}{2}v} = \frac{p_1 \rho}{R} \cdot \frac{\phi + (1-m)}{\phi + \frac{1}{2}}$$

which is the initial pressure O H ($=$ P F $=$ N V) in the low-pressure cylinder. This steam is driven before the high-pressure piston, and drives the low-pressure piston before it till half-stroke, when its volume is $U + \frac{1}{2}V$, and the pressure S K is, therefore,

$$= \frac{p_1 \rho}{R} \cdot \frac{U + v\,(1-m)}{U + \frac{1}{2}V} = \frac{p_1 \rho}{R} \cdot \frac{\phi + (1-m)}{\phi + \frac{1}{2}\lambda}$$

But at this point the high-pressure cylinder, containing a volume v of steam at pressure $\frac{p_1 V}{R v}$, opens to the reservoir, and the pressure becomes

$$= \frac{\frac{p_1 \rho}{R}\left\{U + v\,(1-m)\right\} + v\frac{p_1 V}{R v}}{v + U + \frac{1}{2}V} = \frac{p_1}{R} \cdot \frac{\rho\, U + \rho\, v\,(1-m) + V}{v + U + \frac{1}{2}V}$$

$$= \frac{p_1}{R} \cdot \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{1 + \phi + \frac{1}{2}\lambda} = \text{S L} = \text{R D}.$$

Thus all the points have been obtained, and the diagrams can be drawn.

Fig. 889 has been drawn for cylinders having a ratio of 4 to 1; the steam being cut off at half-stroke in the small cylinder, and at ·55 of the stroke in the large cylinder. The initial pressure O A = 70 lbs., and the condenser pressure N N′ = 8 lbs. per sq. inch. The volume of the reservoir has been taken equal to the volume of the small cylinder.

It will be seen that there is a considerable drop of pressure on the admission to the reservoir, with a corresponding increase in the reservoir pressure, which produces a sudden jump in the low-pressure diagram. In an actual case this jump would be lessened by the effect of the release before the end of the stroke, and of throttling in the passages between the cylinders, and it would appear more in the form of a curve convex to O N.

It will also be seen that a large portion of the work due to expansion is lost, and consequently that the engine is not economical so far as the *theoretical* action of the steam is concerned. This sudden fall of pressure without the performance of work would possibly have the effect of heating the steam somewhat, but there would still be a loss when there is a fall of pressure, as only a percentage of this heat can be converted into mechanical work. If the work due to expansion be fully realised, this drop will become zero and we shall have—

$$\text{R C} = \text{R D}.$$

$$\text{or } \frac{p_1 \text{V}}{\text{R} v} = \frac{p_1}{\text{R}} \cdot \frac{\rho \left\{ \text{U} + v(1-m) \right\} + \text{V}}{v + \text{U} + \frac{1}{2}\text{V}}$$

$$\text{or } \frac{\text{V}}{v} = \lambda = \frac{\rho \left\{ \text{U} + v(1-m) \right\} + \text{V}}{v + \text{U} + \frac{1}{2}\text{V}}$$

$$\frac{\rho}{\lambda} = \frac{v + \text{U} + \frac{1}{2}\text{V}}{\text{U} + v(1-m) + \frac{\text{V}}{\rho}}$$

But $\text{V} = \lambda v$, and $\text{U} = \phi v$. Then, by substitution, we get

$$\frac{\rho}{\lambda} = \frac{1 + \phi + \frac{1}{2}\lambda}{(1-m) + \phi + \frac{\lambda}{\rho}}$$

$$\rho \left\{ (1-m) + \phi \right\} + \lambda = \lambda + \lambda(\phi + \tfrac{1}{2}\lambda) \cdot$$

Solving for λ we get, $\lambda = -\phi \pm \sqrt{\left\{ 2\rho(1-m+\phi) + \phi \right\}}$

Of course the positive sign of the radical must be taken, as λ cannot be negative. From this equation, if ρ and ϕ be given, we can find the value of λ, that would prevent fall of pressure on the admission to the reservoir.

The following table gives a few values of λ, for different points of cut-off, that satisfy the foregoing conditions:—

$\frac{1}{\rho} =$	·5	·55	·6	·65	·7	·75	·8
$\rho =$	2	1·82	1·67	1·54	1·43	1·33	1·25
λ for $\phi = 1$	2	1·88	1·76	1·66	1·57	1·48	1·4
λ for $\phi = 2$	2	1·86	1·74	1·63	1·53	1·44	1·36
λ for $\phi = 3$	2	1·85	1·73	1·61	1·51	1·42	1·33

an intermediate reservoir, the cut-off in the low-pressure cylinder being before half-stroke.—The action of the steam in this case is very similar to that in the last case, with the exception that when the high-pressure cylinder opens to the reservoir the communication with the low-pressure cylinder is closed, so that the increase of pressure takes place only in the reservoir, and tends to increase the initial pressure in the low-pressure cylinder. This is shown in Fig. 390, M being the point corresponding to the release from the high-pressure cylinder, and the steam is compressed to V, N V being the initial pressure in the low-pressure cylinder.

There will be a slightly different value for the quantity $1-m$. In Fig. 891

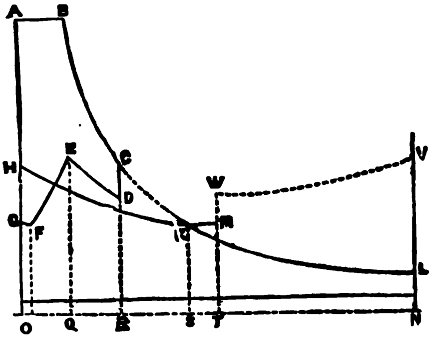


FIG. 390.

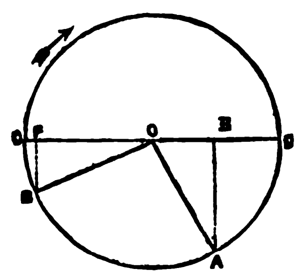


FIG. 391.

let O A be the position of the low-pressure crank at time of cut-off; and O B the position of the high-pressure crank.

$$\frac{\mathrm{DF}}{\mathrm{DC}}=m=\frac{1+\sin\theta}{2};\ \frac{\mathrm{DE}}{\mathrm{DC}}=\frac{1-\cos\theta}{2}=\frac{1}{\rho}$$

$$\therefore\ \cos\theta=\frac{\rho-2}{\rho};$$

$$\text{and } \sin\theta=\sqrt{1-\cos^2\theta}=\frac{2}{\rho}\sqrt{\rho-1}.$$

$$\therefore\ m=\frac{\rho+2\sqrt{\rho-1}}{2\rho}\ \text{ and } 1-m=\frac{\rho-2\sqrt{\rho-1}}{2\rho}.$$

The following table gives the values of $1-m$ for a few values of $\frac{1}{\rho}$.

$\frac{1}{\rho}=$	·2	·25	·3	·35	·4	·45
$1-m=$	·01	0·067	0·043	0·023	0·008	0·003

The final pressure in each of the cylinders is the same as before, viz. in the low-pressure cylinder, N L $=\frac{p_1}{\mathrm{R}}$, and in the high-pressure cylinder, R C $=\frac{p_1}{\mathrm{R}}\cdot\frac{\mathrm{V}}{v}=\frac{p_1}{\mathrm{R}}\lambda$. The pressure at cut-off, S K $=\frac{p_1\rho}{\mathrm{R}}$.

At this instant the volume occupied by the steam is $\mathrm{U}+v\,(1-m)$.

At half-stroke the volume is reduced to U, and the pressure, T M, is therefore $=\frac{p_1\rho}{\mathrm{R}}\cdot\frac{\mathrm{U}+v\,(1-m)}{\mathrm{U}}=\frac{p_1\rho}{\mathrm{R}}\cdot\frac{\phi+(1-m)}{\phi}$.

At this instant the high-pressure cylinder containing a volume, v, of steam at the pressure $\frac{p_1}{R}\frac{V}{v}$ opens to the reservoir, and the pressure becomes

$$\frac{\frac{p_1 \rho}{R}\left\{U + v(1-m)\right\} + v\frac{p_1}{R}\frac{V}{v}}{v + U}$$

$$= \frac{p_1}{R} \cdot \frac{\rho U + \rho v(1-m) + V}{v+U} = \frac{p_1}{R} \cdot \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{1+\phi} = TW = RD.$$

At the end of the stroke of the low-pressure piston this steam occupies the volume $U + \frac{1}{2}v$, and its pressure is therefore,

$$= \frac{p_1}{R} \cdot \frac{\rho U + \rho v(1-m) + V}{U + \frac{1}{2}v} = \frac{p_1}{R} \cdot \frac{\rho\left\{\phi + (1-m)\right\} + \lambda}{\phi + \frac{1}{2}}$$

which is the initial pressure, $OH = NV$, in the low-pressure cylinder. F is the point in the return stroke of the high-pressure piston corresponding to the point of cut-off in the low-pressure cylinder. Consequently the pressure $PF = SK = \frac{p_1 \rho}{R}$. The pressure OG is $= TM$. Thus all the points have been determined and the diagrams can be drawn in any given case.

Fig. 890 has been drawn for the same engine as Fig. 889, the total expansion was also the same, the only difference being that in this case the steam has been cut off at ·4 of the stroke in the low-pressure cylinder instead of ·55 as in the previous example.

There is in this case also a fall of pressure on the admission to the reservoir, with a corresponding increase in the pressure. The drop is, however, not so great nor so injurious, as it tends to increase the initial pressure of the steam in the low-pressure cylinder instead of the pressure at the middle of the stroke. All the work due to the expansion, however, is not realised, and there is still a considerable loss. If there were no drop we should have $RC = RD$,

$$\text{or } \frac{p_1}{R} \cdot \frac{V}{v} = \frac{p_1}{R} \cdot \frac{\rho\left\{U + v(1-m)\right\} + V}{v + U}$$

Putting $V = \lambda v$, and $U = \phi v$, we get $\dfrac{\lambda}{\rho} = \dfrac{(1-m) + \phi + \frac{\lambda}{\rho}}{1+\phi}$

$$\lambda(1+\phi) = \rho\left\{(1-m) + \phi\right\} + \lambda$$

$$\frac{\lambda}{\rho} = \frac{(1-m) + \phi}{\phi}.$$

From this we get the following table:—

$\frac{1}{\rho} =$	·2	·25	·3	·35	·4	·45
$\rho =$	5·0	4·0	3·33	2·86	2·50	2·222
λ for $\phi = 1$	5·5	4·27	3·48	2·92	2·52	2·228
λ for $\phi = 2$	5·25	4·13	3·41	2·89	2·51	2·225
λ for $\phi = 3$	5·17	4·09	3·33	2·88	2·507	2·224

points can be easily found in any given case, as the values of $(1 - m)$ are given in the tables.

On reference to the preceding table, it will be seen that with this type of engine, if the ratio of cylinders is more than two to one, the cut-off in the low-pressure cylinder should be arranged to take place before half-stroke, to prevent loss from sudden expansion, and consequently an expansion valve would be required on the low-pressure cylinder in order to enable the full benefit of the expansion of the steam to be realised.

Three cylinder compound engines with cranks at equal angles.—There are three cases:—

1. When both low-pressure cylinders are open to the reservoir at the time that the high-pressure cylinder exhausts into it; that is, when the cut-off in each low-pressure cylinder is after 0·75 of the stroke.

2. When only one low-pressure cylinder is open to the reservoir when the high-pressure exhausts into it; that is, when the cut-off in each low-pressure cylinder is between 0·25 and 0·75 of the stroke.

3. When neither of the low-pressure cylinders is open to the reservoir when the high-pressure cylinder exhausts into it; that is, when the cut-off in each low-pressure cylinder is before 0·25 of the stroke.

(1) *Cut-off in the low-pressure cylinder after* 0·75 *of the stroke.*

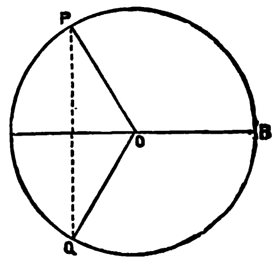


FIG. 392.

By reference to Fig. 392, showing the positions of the several cranks, when the steam is exhausted from the high-pressure cylinder to the reservoir, it will be seen that if OP and OQ represent the low-pressure cranks and OR the high-pressure crank, while one low-pressure cylinder gets steam at one-quarter stroke, the other does not get its supply until the piston has traversed three-fourths of its stroke, and is near the point at which cut-off takes place. The work done by the two low-pressure cylinders would in consequence be very unequal, so that this case would not occur in practice. In all three cylinder compound engines the cut-off in the low-pressure cylinders should be arranged to take place before 0·75 of the stroke.

(2) *Cut-off in the low-pressure cylinders between* 0·25 *and* 0·75 *of the stroke.*

This is the most general case that occurs in practice.

Let v = volume of the high-pressure cylinder.
V = " each low- "
U = " the intermediate reservoir.
R = total ratio of expansion.
r = ratio of expansion in the high-pressure cylinder.
ρ = " " each low- "
λ = ratio of each low-pressure cylinder to the high-pressure cylinder.
ϕ = ratio of the intermediate reservoir to the high-pressure cylinder.

So that $V = \lambda v$; $U = \phi v$; and $R = 2 \lambda r$.

Also, for brevity, let the symbols α and β represent the low-pressure cylinders, and γ the high-pressure cylinder.

In the first place it will be necessary to investigate expressions for the distances of the high-pressure piston from the ends of the stroke when each of the low-pressure pistons are at the points of cut-off.

In Fig. 393 let O P, O Q represent the positions of the cranks of the low-

pressure cylinders α and β respectively, and O R the position of the crank of the high-pressure cylinder γ at the point of cut-off in α.

$$\text{Then } \frac{AC}{AB} = \frac{1}{\rho} = \frac{1 - \cos\theta}{2}$$

$$\text{or } \cos\theta = \frac{\rho - 2}{\rho}$$

$$\sin\theta = \sqrt{1 - \cos^2\theta} = \frac{2}{\rho}\sqrt{\rho - 1}$$

At this point the distance of the high-pressure piston from the end of its stroke is represented by B D, so that the fraction of the high-pressure cylinder open to the reservoir is represented by

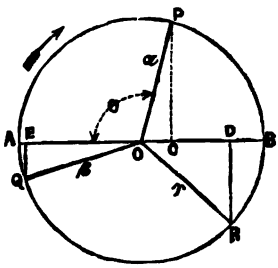


Fig. 893.

$$\frac{AD}{AB} = \frac{1 + \cos(\theta - 60^\circ)}{2}$$

$$= \frac{1 + \frac{1}{2}\cos\theta + \frac{1}{2}\sqrt{3}.\sin\theta}{2}$$

$$= \frac{3\rho + 2\sqrt{3(\rho - 1)} - 2}{4\rho} = l, \text{ say}$$

$$\frac{AE}{AB} = \frac{1 + \cos(\theta + 60^\circ)}{2}$$

$$= \frac{1 + \frac{1}{2}\cos\theta - \frac{1}{2}\sqrt{3}\sin\theta}{2} = \frac{3\rho - 2\sqrt{3(\rho - 1)} - 2}{4\rho}$$

Let this expression be denoted by m.

The following table gives values of l and m for different values of ρ:—

$\frac{1}{\rho}=$	0·25	0·30	0·35	0·40	0·45	0·50	0·55	0·60	0·65	0·70	0·75
$\rho=$	4·0	3·33	2·86	2·50	2·22	2·0	1·82	1·67	1·54	1·43	1·33
$l=$	1·0	·997	·988	·974	·955	·933	·906	·875	·838	·797	·748
$1-l=$	0·0	·003	·012	·026	·045	·067	·094	·125	·162	·203	·252
$m=$	·250	·203	·162	·126	·094	·067	·044	·026	·010	·003	0·00

In investigating the nature of the diagram let us first assume that the cut-off and the final pressures in each of the low-pressure cylinders are the same.

In this case the final pressure in each of the low-pressure cylinders is $= \frac{p_1}{R}$; and the cut-off pressure in each low-pressure cylinder $= \frac{\rho p_1}{R}$.

Let the theoretical diagrams be represented by Figs. 894, 895, and 896; Fig. 896 being from the high-pressure cylinder γ and Figs. 894 and 895 from the low-pressure cylinders α and β respectively. Also let the pressures at the different points be denoted by the letter p, with the suffix corresponding to the number of the point on the diagrams.

First trace the action of the steam in the cylinder α.

The total volume occupied by the steam at the point of cut-off in α is

$$= U + \frac{V}{\rho} + lv$$

and the pressure at this point is $= \frac{\rho p_1}{R} = p_{13}$.

This is also $= p_{16}$ and $= p_5$.

At quarter-stroke of a, *just after* the high-pressure cylinder has exhausted into the reservoir, the volume occupied by the steam is

$$= U + v + \frac{V}{4}$$

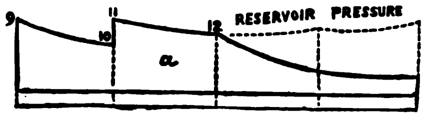


FIG. 394

and the pressure is therefore

$$= \frac{\rho p_1}{R} \cdot \frac{U + \frac{V}{\rho} + lv}{U + \frac{V}{4} + v}$$

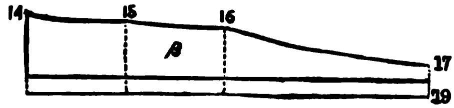


FIG. 395.

$$= \frac{\rho p_1}{R} \cdot \frac{\phi + \frac{\lambda}{\rho} + l}{\phi + \frac{\lambda}{4} + 1} = p_4 = p_{11}.$$

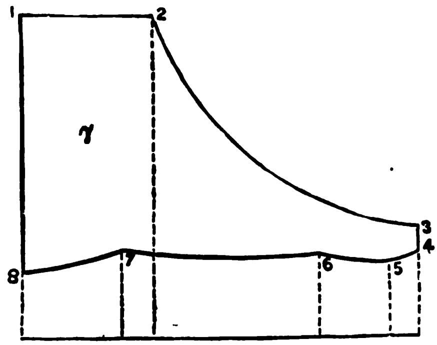


FIG. 396.

The volume occupied by the steam in the reservoir, &c., *immediately before* the high-pressure cylinder exhausts is $= U + \frac{V}{4}$; and its pressure is represented by p_{10}.

At this point a volume, v, of steam at a pressure $\frac{p_1}{r}$ is admitted to the reservoir from the high-pressure cylinder and the pressure rises to p_{11}, the steam then occupying the volume, $U + \frac{V}{4} + v$.

Therefore we have,

$$p_{10}\left(U + \frac{V}{4}\right) + \frac{p_1}{r} v = p_{11}\left(U + \frac{V}{4} + v\right) = \frac{\rho p_1}{R}\left(U + \frac{V}{\rho} + lv\right)$$

$$\text{or } p_{10} = \frac{\frac{\rho p_1}{R}\left(U + \frac{V}{\rho} + lv\right) - \frac{p_1}{r}v}{U + \frac{V}{4}}$$

But $r = \frac{R}{2\lambda}$. Therefore $\frac{p_1}{r} = \frac{2\lambda p_1}{R}$

$$\text{and } p_{10} = \frac{p_1}{R} \cdot \frac{\rho\left(U + lv\right) + V - 2\lambda v}{U + \frac{V}{4}} = \frac{p_1}{R} \cdot \frac{\rho\left(\phi + l\right) - \lambda}{\phi + \frac{\lambda}{4}} = p_8 \text{ also.}$$

After the steam is cut off in a, the steam remaining in the reservoir is compressed behind the high-pressure piston, until the commencement of the stroke of β.

The volume of steam in the reservoir at cut-off of a is $= U + lv$.

At the commencement of the stroke of β this has been altered to $U + \frac{8}{4}v$;

and the pressure is therefore $= \frac{\rho p_1}{R} \cdot \frac{U + l v}{U + \frac{3}{4} v}$

$$= \frac{\rho p_1}{R} \cdot \frac{\phi + l}{\phi + \cdot 75} = \text{initial in } \beta = p_{14} = \text{also } p_6.$$

This steam acts on the piston of β and is acted on by the high-pressure piston till quarter-stroke of β, when a is ready to commence its return stroke. The volume occupied by the steam at this point is $= U + \frac{1}{4}\left(V + v\right)$, and the pressure is, therefore, $= \frac{\rho p_1}{R} \cdot \frac{U + l v}{U + \frac{1}{4}(V + v)}$

$$= \frac{\rho p_1}{R} \cdot \frac{\phi + l}{\phi + \frac{1}{4}(\lambda + 1)} = \text{initial pressure in } a, = p_9 = p_{15} = p_7.$$

Until the point of cut-off in β, this steam acts on the two low-pressure pistons, and is acted on by the high-pressure piston. At the point of cut-off in β, the volume occupied by the steam is

$$= U + \frac{V}{\rho} + (1 - l)\, V + m v$$

and the pressure is, therefore,

$$= \frac{\rho p_1}{R} \cdot \frac{U + l v}{U + \frac{V}{\rho} + \left(1 - l\right) V + m v} = \frac{\rho p_1}{R} \cdot \frac{\phi + l}{\phi + \lambda\left(\frac{1}{\rho} + 1 - l\right) + m}$$

$= p_{16}$, the pressure at the point of cut-off in β.

By the assumption previously made, viz. that the pressures at the points of cut-off in the cylinders a and β are to be the same, this must be equal to $\frac{\rho p_1}{R}$.

$$\text{Therefore, } \frac{\phi + l}{\phi + \lambda\left(\frac{1}{\rho} + 1 - l\right) + m} \text{ must be } = 1$$

$$\text{or } \phi + \lambda\left(\frac{1}{\rho} + 1 - l\right) + m = \phi + l \quad \therefore \quad \lambda = \frac{l - m}{\frac{1}{\rho} + 1 - l}$$

From this we see that only when a certain relation exists between the ratios of cylinders and the point of cut-off in the low-pressure cylinders, can the final pressures and rates of expansion be the same in each of the low-pressure cylinders.

From the foregoing equation the necessary ratios of cylinders for certain points of cut-off, in order to make the final pressures in the low-pressure cylinders the same, can be readily found, and a few values are given below:—

$\frac{1}{\rho} =$	0·25	0·30	0·35	0·40	0·45	0·50	0·55	0·60	0·65	0·70	0·75
$l =$	4·0	3·33	2·86	2·50	2·22	2·0	1·82	1·67	1·54	1·43	1·33
$\lambda =$	3·0	2·62	2·28	1·99	1·75	1·52	1·34	1·17	1·02	0·88	0·75

From this table it appears that in the majority of cases in general practice, the expansion and cut-off will not be exactly the same in the two low-pressure cylinders. In general, if the steam is cut off at the same part of the stroke in each of the two low-pressure cylinders, the final pressures

will be different.

First assume the final pressures the same in each of the low-pressure cylinders, but the cut-offs different.

Let ρ represent the ratio of expansion in a.

" ρ_1 " " β.

The same method of reasoning must be applied as in the previous case, and it will be found that the only pressure that is altered is p_{16}.

This becomes

$$p_{16} = \frac{\rho p_1}{\mathrm{R}} \cdot \frac{\phi + l}{\phi + \lambda\left(\dfrac{1}{\rho_1} + 1 - l_1\right) + m_1}$$

where l_1 and m_1 are the values of l and m corresponding to the ratio of expansion ρ_1.

The final pressure in β, p_{17}, is, therefore,

$$= \frac{\rho}{\rho_1} \cdot \frac{p_1}{\mathrm{R}} \cdot \frac{\phi + l}{\phi + \lambda\left(\dfrac{1}{\rho_1} + 1 - l_1\right) + m_1}$$

But by supposition this is equal to the final pressure in a,

$$\text{or } \frac{p_1}{\mathrm{R}} = \frac{\rho}{\rho_1} \cdot \frac{p_1}{\mathrm{R}} \cdot \frac{\phi + l}{\phi + \lambda\left(\dfrac{1}{\rho_1} + 1 - l_1\right) + m_1}$$

$$\text{or } \frac{\rho}{\rho_1} = \frac{\phi + \lambda\left(\dfrac{1}{\rho_1} + 1 - l_1\right) + m_1}{\phi + l}$$

$$\rho(\phi + l) = \rho_1\left\{\phi + \lambda(1 - l_1) + m_1\right\} + \lambda$$

$$\text{or } \rho_1 = \frac{\rho(\phi + l) - \lambda}{\lambda(1 - l_1) + \phi + m_1}.$$

As ρ and ρ_1 are not very different, it will be sufficiently accurate for practical purposes to take l_1 and m_1, the same as l and m. In this case the value of ρ_1 for any given value of ρ can be obtained.

Secondly.—Assume that the cut-off is the same in each of the low-pressure cylinders, so that, generally, the final pressures would be different.

Let p_a represent the final pressure in a.

" p_β " " β.

We shall be sufficiently accurate if we assume that $\frac{p_1}{\mathrm{R}}$, which is the pressure due to the total expansion, is a mean between p_a and p_β.

$$\text{or, } \tfrac{1}{2}\left(p_a + p_\beta\right) = \frac{p_1}{\mathrm{R}}$$

The expressions for the pressures at each of the points will be exactly similar to those previously given, with the exception that p_a must be substituted for $\frac{p_1}{\mathrm{R}}$.

The expression for the pressure at the point of cut-off in β is

$$p_{16} = \rho p_a \cdot \frac{\phi + l}{\phi + \lambda\left(\dfrac{1}{\rho} + 1 - l\right) + m} = \rho p_\beta \text{ by hypothesis.}$$

$$\text{Therefore } \frac{p_a}{p_\beta} = \frac{\phi + \lambda\left(\dfrac{1}{\rho} + 1 - l\right) + m}{\phi + l}.$$

From this equation the ratios of the final pressures in the two low-pressure cylinders to each other can be found; and then from the equation,

$$\tfrac{1}{2}\left(p_{\alpha}+p_{\beta}\right)=\frac{p_1}{R}$$

the actual final pressures may be obtained.

(8) *Cut-off in low-pressure cylinders before quarter-stroke.*

In this case the receiver is never in communication with more than one low-pressure cylinder at the same time.

This case would seldom occur in practice except possibly when the engines were being worked at reduced power. In engines in which the combined volume of the low-pressure cylinders is more than four times that of the high-pressure cylinder, loss from sudden expansion on admission to the receiver can only be avoided by cutting off the admission to the low-pressure cylinders before quarter-stroke, but practical considerations do not admit of this being done, so that the case is of little practical interest, and is not dealt with further. It will be a useful exercise, however, for the student to go through the investigation and ascertain the form of the diagrams.

Triple-expansion engines with cranks at 120°, and cut-off in cylinders between 0·25 and 0·75 of stroke.

(*a*) First *with LP crank leading*:—

Let v_1, v_2, and v_3 be the volumes of HP, MP, and LP cylinders.

w_1 and w_2 = volumes of 1st and 2nd reservoirs

r_1, r_2, and r_3 = ratios of expansion in HP, MP, and LP cyls.

$$\lambda_1=\frac{\text{MP cyl.}}{\text{HP cyl.}}\quad\text{and } \lambda_2=\frac{\text{LP cyl.}}{\text{HP cyl.}}$$

R = total ratio of expansion.

Then $R=\dfrac{v_3\,r_1}{v_1}=\lambda_2 r_1$; $v_2=\lambda_1 v_1$; and $v_3=\lambda_2 v_1$.

Let $w_1=\phi_1 v_1$ and $w_2=\phi_2 v_2$

Let O P (Fig. 397) be the position of MP crank at cut-off in the MP cylinder,

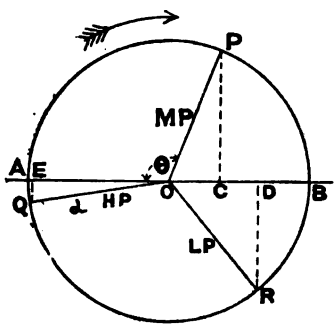


Fig. 397.

$$\text{then } \frac{AC}{AB}=\frac{1}{r_2}=\frac{1-\cos\theta}{2}$$

$$\text{therefore } \cos\theta=\frac{r_2-2}{r_2}$$

$$\text{and } \sin\theta=\sqrt{1-\cos^2\theta}=\frac{2}{r_2}\sqrt{r_2-1}$$

The distance of the HP piston from the end of the stroke is = A E, so the total volume of steam is $w_1+\dfrac{AE}{AB}$ (HP cyl.) $+\dfrac{AC}{AB}$ (MP cyl.)

$$\frac{AE}{AB}=\frac{1+\cos(\theta+60^\circ)}{2}=\frac{-\frac{1}{2}\sqrt{3}\sin\theta+\frac{1}{2}\cos\theta+1}{2}=\frac{3r_2-2\sqrt{3(r_2-1)}-2}{4r_2}=m$$

say.

Table for values of m will be found on p. 468. We can now determine the various cut-off and terminal pressures in the three cylinders.

Thus, if p_1 be initial pressure in HP cyl.

Cut-off pressure in HP cyl. = p_1

Terminal „ „ HP „ $=\dfrac{p_1}{r_1}$

Cut-off pressure in MP cyl. $= \dfrac{\quad}{\lambda_1 r_1}$

Terminal " " MP " $= \dfrac{p_1 v_1}{r_1 v_2} = \dfrac{p_1}{\lambda_1 r_1}$

Cut-off " " LP " $= \dfrac{r_3 p_1}{\lambda_2 r_1}$

Terminal " " LP " $= \dfrac{p_1}{\lambda_2 r_1}$

Let the theoretical diagram be as shown in fig. 398.

First trace the action of the steam in the MP cylinder and exhaust from the HP cylinder.

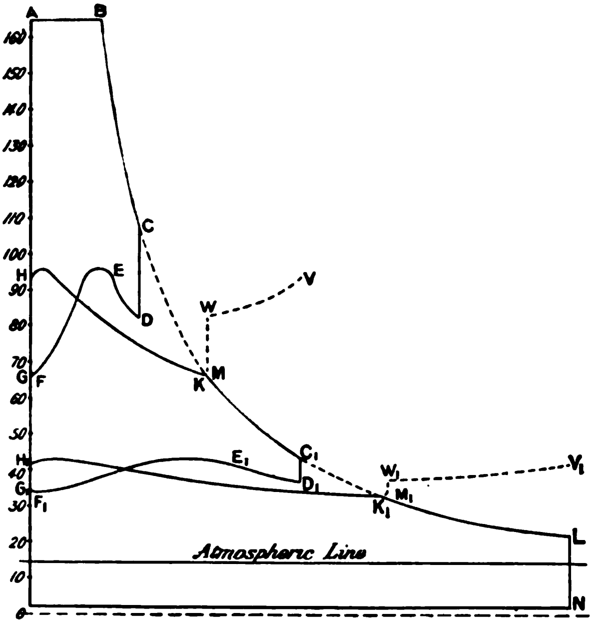


FIG. 398.

The total volume occupied by the steam at the point of cut-off in the MP cylinder is $w_1 + \dfrac{v_2}{r_2} + mv_1$, and the pressure is $\dfrac{r_3}{r_1}\dfrac{p_1}{\lambda_1}$.

This is $p_K = p_F$.

At 0·75 stroke of the MP cylinder, just after the HP cylinder has exhausted into the reservoir, the volume occupied by the same steam is $w_1 + v_1$ and for the pressure p_D we have $p_D\ (w_1 + v_1) = p_K\ (m\,v_1 + w_1) + p_C \times v_1$

$$p_D = \frac{\dfrac{r_2 p_1}{r_1 \lambda_1}\left(mv_1 + w_1 + \dfrac{\lambda_1}{r_2} v_1\right)}{w_1 + v_1} = p_W$$

This steam is now compressed by HP piston until at E the MP opens to steam. The pressure at E, p_E is given by

$$p_E\ (\cdot75\ v_1 + w_1) = p_D\ (v_1 + w_1)$$

$$p_E = p_D \frac{v_1 + w_1}{\cdot75\ v_1 + w_1}$$

As the MP piston moves slowly while the HP piston moves quickly the compression still goes on until $\lambda_1 \times$ (velocity of MP piston) = velocity of HP piston (i.e. until the volume of steam contained in the receiver and between the two pistons begins to increase).

Then expansion takes place, as the MP piston sweeps out a greater volume than the HP, until at F the steam is cut off in MP cylinder. We then have just before cut-off in MP cylinder, volume $_F = \left(\frac{v_2}{r_2} + w_1 + m\,v_1\right)$ and p_F = cut-off pressure in MP.

For the remainder of the HP stroke the steam is compressed by the HP piston until just before admission, when we have

$$p_G\,w_1 = p_F\,(m\,v_1 + w_1) \;\therefore\; p_G = p_F\,\frac{(m\,v_1 + w_1)}{w_1}$$

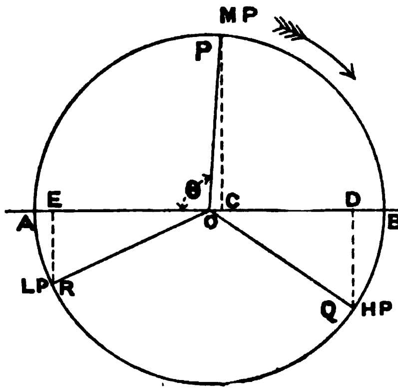


FIG. 399.

We have now all the points on the exhaust line of HP diagram and on the steam and reservoir pressure line of MP diagram.

The LP steam line and MP exhaust is obtained in exactly the same way.

Fig. 398 shows the diagrams obtained by these means with cut-off in each cylinder at 0·65 of stroke, the cylinder ratios being as 1 : 2¼ : 5, the volumes of the receivers each being equal to 1½ time the volume of the preceding cylinder and the initial pressure 165 lbs. absolute.

(*b*) Next consider the same engine but *with HP crank leading*.

Suppose in Fig. 399 the MP to be on the point of cut-off, then

$$\frac{AC}{AB} = \frac{1}{r_2} = \frac{1-\cos\theta}{2} \;\therefore\; \cos\theta = \frac{r_2-2}{r_2}$$

$$\text{and } \frac{AD}{AB} = \frac{1+\cos\overline{\theta-60}^\circ}{2} = \frac{1+\frac{1}{2}\cos\theta + \frac{1}{2}\sqrt{3}\,.\,\sin\theta}{2}$$

$$= \frac{3\,r_2 + 2\sqrt{3\,(r_2-1)} - 2}{4\,r_2} = l \text{ say.}$$

Then the cut-off and terminal pressures for the different points can be obtained as before with the LP crank leading, the final volumes in each case being the same.

First consider the exhaust line of the HP diagram and the MP steam-line (Fig. 400).

At cut-off in MP diagram the volume is $\frac{v_2}{r_2} + l\,v_1 + w_1$, and pressure at cut-off $= p_M = \frac{r_2\,p_1}{\lambda_1\,r_1} = p_E$, E being where the HP piston has travelled AB − AD on the return stroke, or $(1-l)$ stroke.

This steam is now compressed by the HP piston until the MP again opens to steam.

We then have at F (just before the MP opens to steam)

$$p_F\,(\cdot 25\,v_1 + w_1) = p_E\,(w_1 + l\,v_1)$$

$$\therefore\; p_F = \frac{w_1 + l\,v_1}{\cdot 25\,v_1 + w_1}\,p_E.$$

the increase of volume due to steam admitted to MP cylinder becomes equal to the decrease due to the motion of the HP piston or until

$$\frac{v_2}{v_1} \times \text{velocity of MP piston} = \text{velocity of HP piston, or}$$

$$\lambda_1 \times \text{velocity of MP piston} = \text{velocity of HP piston.}$$

At end of HP stroke just before exhaust the volume of steam is

$$\cdot 25\, v_2 + w_1 \text{ and its pressure} = p_G$$

$$\therefore\ p_G\,(\cdot 25\, v_2 + w_1) = p_E\,(w_1 + l\, v_1) \text{ and } p_G = \frac{w_1 + l\, v_1}{\cdot 25\, v_2 + w_1}\, p_E.$$

To find the pressure directly after exhaust by the HP cylinder we have

$$p_C\, v_1 + p_G\,(\cdot 25\, v_2 + w_1) = p_D\,(v_1 + \cdot 25 v_2 + w_1)$$

or

$$p_D = \frac{p_C\, v_1 + p_G\,(\cdot 25\, v_2 + w_1)}{v_1 + \cdot 25\, v_2 + w_1}.$$

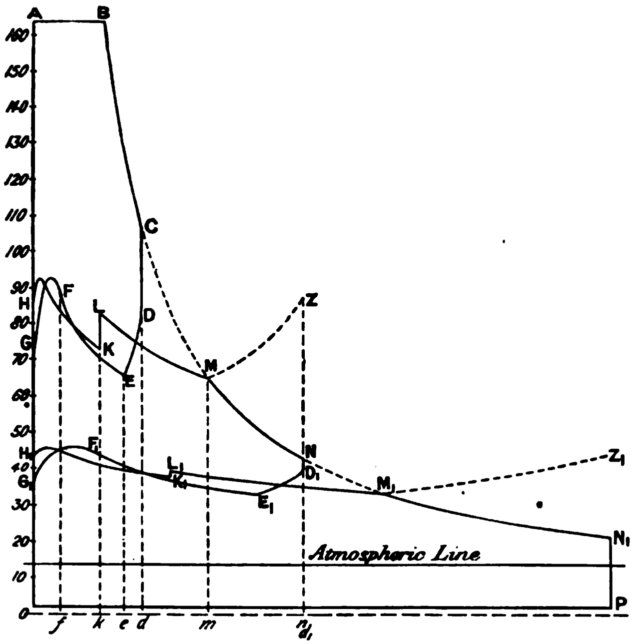


FIG. 400.

The initial pressure in MP cylinder p_H is $= p_F$.

All the pressures have now been found for exhaust line of HP, and this gives the pressures on the steam line of MP diagram and reservoir pressure.

After cut-off in MP cylinder the steam is compressed and not in contact with MP piston until $p_Z = p_H$.

The positions of the points E, L, K, &c., must now be determined. Thus, considering points D and E, $d\,e$ is the distance moved through by the HP piston while the MP moves from angle of 60° with line of centres to the point of cut-off, or if θ_C is the angle of cut-off $\left(\frac{1 - \cos(\theta_C - 60)}{2}\right)$ = fraction of stroke moved through by HP piston

$$= \frac{\text{B D}}{\text{A B}} \times \text{stroke} = (1 - l)\text{ stroke}$$

Again, $e f$ is the distance moved through by the HP piston while the MP moves from cut-off to release

$= (\cdot 75 - 1 + l)$ stroke $= (l - \cdot 25)$ stroke. Also from F to G $= \cdot 25$ stroke.

Considering the steam line of MP diagram, the MP piston moves through $60°$ while pressure alters from p_H to p_K, therefore $o\,k = \cdot 25$ stroke.

Hence for the part from L to M, $k\,m = \left(\frac{1}{r_2} - \cdot 25\right)$ stroke

and for the expansion line M N, $m\,n = \left(1 - \frac{1}{r_2}\right)$ stroke.

The exhaust line of the MP and steam line of the LP are obtained in exactly the same way, the only difference being alterations in r_1, r_2, w_1, &c., to the corresponding value r_2, r_3, w_2, &c.

Fig. 400 shows the diagram calculated for the same engine and proportions of cut-off, &c., as in the previous case.

Four cylinder triple-expansion engine with two LP cranks at right angles (i.e. engine with four cranks at right angles) and cut-off after half-stroke in each cylinder.—This, as regards cut-off, is the general case in practice.

Let P_1 = initial pressure in HP cylinder.

v_1, v_2, and v_3 the volumes of HP, MP, and one LP cylinder.

w_1 and w_2 the volumes of the two reservoirs.

r_1, r_2, and r_3, ratios of expansion in the three cylinders.

$$\lambda_1 = \frac{\text{MP cylinder}}{\text{HP cylinder}} = \frac{v_2}{v_1}.$$

$$\lambda_2 = \frac{\text{one LP cylinder}}{\text{MP cylinder}} = \frac{v_3}{v_2}.$$

ϕ_1 and ϕ_2 = ratios of reservoirs to HP cylinder and MP cylinder.

or $w_1 = \phi_1 v_1$, and $w_2 = \phi_2 v_2$.

R = total ratio of expansion.

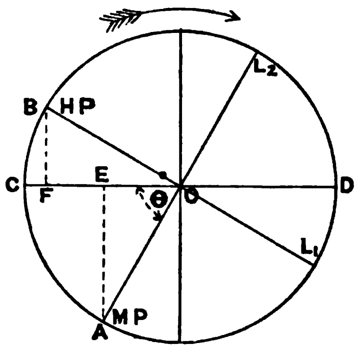


Fig. 401.

In the first place it will be well to notice that only two consecutive cylinders are considered at the same time (in the case of LP steam lines we consider the MP and both LP). Thus first consider the exhaust line of the HP diagram and steam line of the MP diagram.

To find an expression for the distance of the HP piston from the end of the stroke when steam is cut-off in the MP cylinder, let O A be the position of the crank of the MP cylinder at cut-off; O B the corresponding position of HP crank (Fig. 401). Then $\frac{\text{D E}}{\text{D C}} = \frac{1}{r_2}$ and $\frac{\text{C F}}{\text{D C}}$ = fraction of stroke performed by the HP piston when steam is cut off in MP piston $= \frac{1 - \sin\theta}{2}$;

$$\frac{\text{D E}}{\text{D C}} = \frac{1}{r_2} = \frac{1 + \cos\theta}{2} \therefore \cos\theta = \frac{2 - r_2}{r_2}$$

and $\sin\theta = \sqrt{1 - \cos^2\theta} = \frac{2}{r_2}\sqrt{r_2 - 1}$

$$\therefore \frac{\text{C F}}{\text{D C}} = \frac{r_2 - 2\sqrt{r_2 - 1}}{2r_2} = m \text{ say.}$$

Hence the amount of the HP cylinder which is filled with steam at the

equal to $(1-m)$.

Table of values of $1-m$ will be found on p. 462.

In Fig. 402 O A represents the initial pressure of steam in the HP cylinder. At B steam is cut off and expands to C at end of stroke of HP piston and then communication is opened to the reservoir and the pressure falls to the point D where pressure $= p_D$. After this the steam expands in the reservoir and MP cylinder until it is cut off in the latter. This part of the action of the steam is represented by the curve D E; p_E being the pressure when this cut-

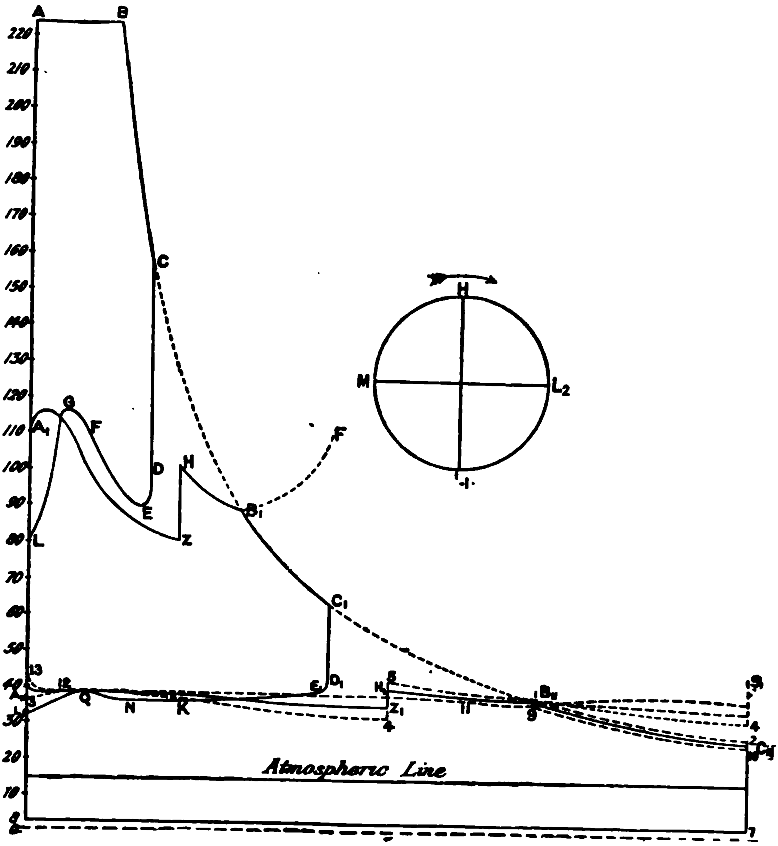


FIG. 402.

off takes place. From this point the steam is compressed behind the HP piston until it has completed its half-stroke at F and pressure $= p_F$ when the MP opens to steam. The MP piston, however, though larger than the HP piston, moves much slower than the HP, and hence the total volume of the steam between the two pistons decreases in volume and, therefore, increases in pressure until the volume becomes a minimum. This is evidently when the rate of increase of volume due to motion of MP piston is equal to the decrease of volume due to motion of HP piston. After this the steam expands and pressure falls to the point L when the pressure is p_L.

The steam line of the MP diagram is deduced from this back-pressure line of HP cylinder. The initial pressure $= p_{A_1}$ is, of course, equal to the pressure at the middle of the back-pressure line of the HP cylinder, i.e. $= p_F$. The pressure now follows the back-pressure line of the HP, rising to a maximum $= p_G$ and then falling to a pressure equal to p_L at half-stroke. The HP now exhausts into the reservoir and the pressure rises to H, where $p_H = p_D$, and then again falls to B_1, due to expansion, the point of cut-off in the MP cylinder, p_{B_1} being equal to p_B. From B_1 the steam expands to C_1 in the MP cylinder, while the reservoir pressure rises to F due to compression by the HP piston, p_F of course being the initial pressure of the MP cylinder $= P_{A_1}$.

Now consider the MP exhaust line and the steam lines of the two low-pressure cylinders. For convenience in obtaining the mean diagram, the two low-pressure diagrams are drawn to twice the volume scale of the MP and the HP diagrams, but to the same pressure scale.

When the MP cylinder opens to exhaust the pressure falls to D_1, where the pressure is $= p_{D_1}$. After this the steam expands to E_1, where one LP cuts off, as till then both the LP cylinders are open to steam. The two LP cylinders may be called L_1P and L_2P for shortness. Then after E_1 the reservoir pressure still falls, as L_2P is open to steam, to K when L_1P cylinder opens to steam, and the pressure falls more rapidly to N when the pressure is the same as at cut-off in the L_2P cylinder. The steam in the reservoir is now compressed by the MP piston, and expanded behind the L_1P piston, and hence as the MP piston is moving quicker than the L_1P, the pressure rises to a maximum at Q, and then falls to L_1 owing to expansion in L_1P cylinder. The steam line of the two LP diagrams are obtained from this MP exhaust line.

Thus the initial pressure of L_2P diagram at 18 is equal to p_{D_1}, and falls to cut-off point 9 in the same way that the pressure falls from D_1 to K. The steam in L_2P cylinder then expands to 10, and opens to exhaust the pressure falling to the constant back-pressure line 7.8.

The initial pressure of the L_1P diagram is at $8 = p_8$, and is equal to the pressure at half-stroke in the exhaust of the MP diagram, and steam line of L_2P diagram. Almost directly after L_1P opens to steam L_2P cylinder cuts off and then the pressure rises in the same way as from N to Q, and then falls to 4 at half-stroke. The MP now opens to exhaust into the reservoir, and the pressure rises to $5 = p_{D_1}$ and then again falls to 1, where steam is cut off, and expands in the cylinder, and then exhausts to the condenser.

The reservoir pressures are also shown. We can now obtain algebraical expressions for the pressures at the different points in order that diagrams may be drawn in any given case.

Since the ratio of expansion in HP cylinder is $= r_1$, the final pressure in HP cylinder $= p_0 = \frac{P_1}{r_1}$, and final pressure in MP cylinder is $= \frac{P_1}{r_1 \lambda_1}$, therefore the cut-off pressure in the MP cylinder is $= \frac{P_1 r_2}{r_1 \lambda_1}$. This is also equal to the pressure in the reservoir at E, and we have therefore steam at the pressure $\frac{P_1 r_2}{\lambda_1 r_1}$ occupying a volume $w_1 + v_1(1-m)$. This steam is compressed behind the HP piston until the beginning of the next stroke of the MP piston, when its volume has been reduced to $w_1 + \frac{v_1}{2}$, and its pressure increased to

$$p_F = \frac{\frac{P_1 r_2}{\lambda_1 r_1}\left(w_1 + v_1(1-m)\right)}{w_1 + \frac{v_1}{2}} = \frac{P_1 r_2}{\lambda_1 r_1}\left(\frac{\phi_1 + 1 - m}{\phi_1 + \frac{1}{2}}\right)$$

which is the initial pressure of MP diagram $= p_{A_1}$. The steam is now driven

than the MP the pressure is increased until the volume begins to de-crease. This it will do when the $\frac{\text{vel. of MP piston}}{\text{vel. of HP piston}} = \frac{\text{HP piston area}}{\text{MP piston area}}$, or if a = angle that the MP crank makes with the line of centres;

when $\lambda_1 \times v_o \sin a = v_o \cos a$, or when $\cot a = \lambda_1$.

The volume now occupied by the steam is $\frac{1}{2} v_2 \operatorname{vers} a + \frac{1}{2} v_1 \operatorname{vers} (90 - a) + w_1$

and its pressure is given by $p_G = \dfrac{\frac{P_1 r_2}{\lambda_1 r_1}(\phi_1 v_1 + \overline{1 - m}\, v_1)}{\frac{1}{2} v_2 \operatorname{vers} a + \frac{1}{2} v_1 \operatorname{vers} (90 - a) + w_1}$

After this the steam expands until its volume is $w_1 + \frac{v_2}{2}$

∴ pressure at L = pressure at Z = p_L.

$$= \frac{p_F \left(w_1 + \frac{v_1}{2}\right)}{w_1 + \frac{v_2}{2}} = \frac{P_1 r_2}{\lambda_1 r_1} \; \frac{(\phi_1 + 1 - m)}{\left(\phi_1 + \frac{\lambda_1}{2}\right)}$$

At this point the HP cylinder opens to exhaust, and we have an HP cylinder full of steam at pressure = p_C admitted to the reservoir, and the final pressure p_D is given by

$$p_D = \frac{\frac{P_1 r_2}{\lambda_1 r_1}\left\{w_1 + v_1(1 - m)\right\} + v_1 \frac{P_1}{r_1}}{v_1 + w_1 + \frac{v_2}{2}}$$

$$= \frac{\frac{P_1 r_2}{\lambda_1 r_1}\left\{\phi_1 + 1 - m\right\} + \frac{P_1}{r_1}}{\left(1 + \frac{\lambda_1}{2} + \phi_1\right)} = \frac{P_1}{\lambda_1 r_1} \cdot \frac{r_2(\phi_1 + 1 - m) + \lambda_1}{1 + \frac{\lambda_1}{2} + \phi_1}$$

Thus all points for the first reservoir are determined.

The calculations for the other diagrams are similar to the above, remembering that now there are two steam lines of the two LP engines and one back-pressure line of MP diagram to be considered in conjunction.

We now want an expression for $\frac{ND}{CD} = \frac{CK}{CD}$ when L_1P is at point of cut-off (Fig. 408), and this is evidently the same expression as found before for the MP cut-off = l say.

$\therefore l = \frac{r_3 - 2\sqrt{r_3} - 1}{2 r_3}$ and $\frac{DK}{DC} = 1 - l$.

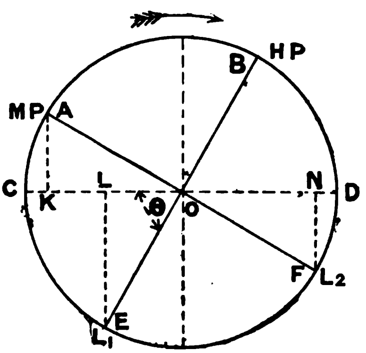


FIG. 408.

Let p_{D_1} be the pressure at D_1 in the second reservoir directly after exhaust from the MP cylinder, the volume of steam is

$$= v_2 + w_2 + \frac{v_3}{2} = v_2\left(1 + \phi_2 + \frac{\lambda_2}{2}\right)$$

This expands till cut-off in L_1P when volume is

$$= \frac{v_3}{r_3} + l\, v_3 + (1 - l)\, v_2 + w_2 = v_2\left(\frac{\lambda_2}{r_3} + l\lambda_2 + (1 - l) + \phi_2\right)$$

And when L_1P is just opening to steam the volume of steam is

$$= \left(\frac{v_2}{2} + \frac{v_2}{2} + w_2\right) = v_2\left(\tfrac{1}{2} + \frac{\lambda_2}{2} + \phi_2\right)$$

Hence the pressure at cut-off in L_1P

$$= p_{D1} \times \frac{1 + \phi_2 + \frac{\lambda_2}{2}}{\frac{\lambda_2}{r_2} + l\lambda_2 + 1 - l + \phi_2} = \text{P say.}$$

The volume $\frac{v_2}{r_2}$ is now cut off from the reservoir in the cylinder L_1P, and we get the initial pressure in L_1P

$$= \frac{p_{D1} \times \left(1 + \phi_2 + \frac{\lambda_2}{2}\right)v_2 - \text{P} \times \frac{v_2}{r_2}}{v_2\left(\tfrac{1}{2} + \frac{\lambda_2}{2} + \phi_2\right)} = \text{P}_3 \text{ say.}$$

At cut-off in L_2P we have this same steam in a volume

$$= lv_2 + \frac{v_2}{r_2} + w_2 + \left(1 - \frac{1}{r_2}\right)v_2.$$

Hence cut-off pressure in L_2P is given by

$$\text{P}_3 \times \frac{v_2\left(\tfrac{1}{2} + \frac{\lambda_2}{2} + \phi_2\right)}{lv_2 + \frac{v_2}{r_2} + w_2 + \left(1 - \frac{1}{r_2}\right)v_2} = \text{P}_2 \text{ say,}$$

and $\frac{\text{P}}{r_2}$ = terminal pressure of L_1, also $\frac{\text{P}_2}{r_2}$ = terminal pressure of L_2.

We can now obtain the cut-off pressures in L_1P and L_2P in terms of the one unknown quantity, p_{D1}.

The weight of steam entering and leaving the engine is the same in one revolution. And this weight of steam is proportional to $p\,v$.

Let $\text{K}p\,v$ = the weight per revolution,

$$\therefore \text{K}\left(\text{P}\frac{v_2}{r_2} + \text{P}_2\frac{v_2}{r_2}\right) = \text{weight passed to condenser,}$$

and $\text{K}\,\text{P}_1\frac{v_1}{r_1}$ = weight entering engine. $\therefore \text{P}\frac{v_2}{r_2} + \text{P}_2\frac{v_2}{r_2} = \text{P}_1\frac{v_1}{r_1}$,

which gives an equation for $\text{P} + \text{P}_2$, and knowing P and P_2 in terms of p_{D1} we can find p_{D1}, and thence P, P_2, and P_3. We have thus all the principal pressures. Thus p_{D1} being known, the pressure falls to E_1 = cut-off pressure of L_1P, and then still falls to K, where pressure = initial pressure of $L_1 = \text{P}_3$, and then falls still quicker to N, when L_2P cuts off. As the MP piston now moves faster than the L_1P piston, we get a rise of pressure to Q, which is obtained as before by equating the rate of diminution of volume due to the MP piston to the rate of increase of volume due to the L_1P piston. The pressure then falls to L_{1}, when the MP again exhausts and the pressure at which point is equal to P_4.

It is easy then to fill in the steam curves for the two LP diagrams. Thus: 3, 4, 5, 1, 2, is the steam-line for the L_1P cylinder, and 13, 12, 11, 9, 10, is the steam line for the L_2P cylinder, found in exactly the same way as described for the steam line of the MP diagram. The mean LP diagram is the mean of these two, viz. A_{11}, Z_1, H_1, B_{11}, C_{11}. The back-pressure line is 7.8.

Four cylinder triple-expansion engine with two LP cranks opposite one another.—The next case to be considered is that of the same engine as in

the previous case, with the cranks differently arranged, that is, having the intermediate crank opposite the high pressure and the two low pressures opposite each other, and at right angles to the other cranks, which is the usual arrangement. The two low-pressure cylinders therefore take steam and exhaust together. The diagram lines between the high pressure and the intermediate are evidently the same as the second case considered in this chapter, viz. a compound engine with cranks at 180° and with an intermediate receiver, and will be as shown at F E D C and G H K in Fig. 387. The diagram for the exhaust line of intermediate pressure and steam line of low pressure is the same as another of our previously investigated cases, viz. that of a compound engine with cranks at right angles, and will therefore be similar to G F E D C and H K L W M of Fig. 389, if, as is usually the case, the cut-off in the low pressure is after half-stroke. The diagram for this case can therefore be easily drawn from previous investigations.

In all these investigations the effect of clearance, always considerable, has been neglected for shortness and simplicity, but it is easily taken account of when the principles worked out above are understood, and it will be a useful exercise for the student to draw the last two diagrams, making the necessary allowances for cylinder clearance and compression.

APPENDIX

(A.) *APPLICATION OF THE INDICATOR DIAGRAM TO DETERMINE THE STRESSES ON CRANK-SHAFTS. CURVES OF TWISTING MOMENTS.*

FOR the sake of simplicity, suppose the obliquity of the connecting-rod and the weights of the reciprocating parts to be neglected.

Let Fig. 404 represent the crank-circle, O being the centre of the shaft, and suppose the forward pressure to be constant throughout the stroke, and equal to P. Then, it is clear that when the crank is in any position, O C, making an angle θ with the line of dead points, the twisting moment exerted will be $= P \times L \sin \theta$; where L represents the length of the crank. If, therefore, the crank-circle be divided into any number of equal parts, each subtending an angle a, the successive twisting moments will be, $P L \sin a$, $P L \sin 2a$, $PL \sin 3a$, &c.

The base line of the diagram of twisting moments is taken to represent the circumference of the crank-circle, and is divided into a number of equal parts representing equal angles of the crank with the line of dead points.

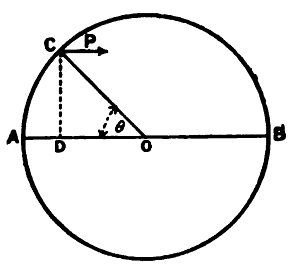


FIG. 404.

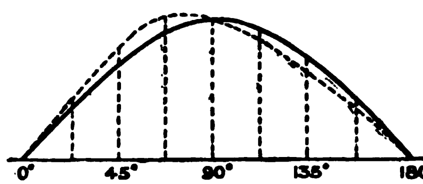


FIG. 405.

Ordinates are set up at these divisions equal to the twisting moment ($P \times L \sin \theta$), for the corresponding angle, and a fair curve is drawn through the ends of the ordinates thus obtained.

In the case under consideration, the curve will be symmetrical, the obliquity of the connecting-rod having been neglected, so that the piston is supposed to have exact harmonic motion. This *curve of sines* is shown by the *full lines* in Fig. 405.

When the obliquity of the connecting-rod is taken into account the difference of speed of piston at the opposite ends of the stroke will be found to destroy the symmetry of the curve of twisting moments. By reference to Fig. 406, it will be seen that, if ϕ be the angle of the connecting-rod when the crank makes an angle θ with the line of dead points, the twisting moment, instead of being $= P \times L \sin \theta$ simply, is

$$= Q \times O\,C \sin O\,C\,D = Q \times L \sin (\theta + \phi)$$

$$\text{where } Q = \text{thrust on the connecting-rod} = \frac{P}{\cos \phi}$$

$$\therefore \text{ the twisting moment is } P \times L \frac{\sin (\theta + \phi)}{\cos \phi} = P.L (\sin \theta + \cos \theta \tan \phi)$$

The second term in the bracket is always small, and when θ is greater than 90°, will be negative: so that it is clear that the curve will be fuller in the first quarter and less in the second quarter revolution, as shown by the dotted lines in Fig. 405.

If the crank-circle be divided into sixteen equal parts, each subtending an angle of $22\frac{1}{2}°$, the successive multipliers of P.L will be

	1	2	3	4	5	6	7	8	9
Connecting-rod, infinite	0	0·383	0·707	0·924	1·0	0·924	0·707	0·383	0
Connecting-rod = four cranks	0	0·428	0·834	1·015	1·0	0·833	0·580	0·338	0

In practice it is generally convenient to take P in tons, and L in inches, so that the resulting twisting moments are obtained in inch-tons.

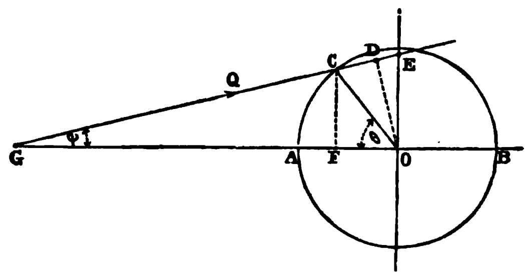


FIG. 406.

We now proceed to explain how actual indicator diagrams taken from the cylinders of an engine may be utilised to show the variations of the twisting moments on the crank-shaft throughout the stroke. The pressure on the piston will in this case vary at each point of the stroke, instead of being constant as before assumed, and the curve of twisting moments will be much less regular, and will fall very rapidly from the point at which expansion begins.

The base line of the diagram of twisting moments is divided into equal parts as before to represent the successive angles of the crank. Ordinates are drawn across the indicator diagram at the points corresponding to the respective positions of the piston for the several angles of the crank, which will give the pressure of steam per square inch on the piston for the given angle of the crank. This pressure multiplied by the area of the piston, by the length of the crank, and by the quantity represented by $\dfrac{\sin(\theta+\phi)}{\cos\phi}$ will give the corresponding twisting moment on the crank-shaft, or if

p = pressure measured on the diagram, in pounds.
A = area of piston, in square inches.
L = length of crank, in inches,

the twisting moment in inch-tons, exerted when the crank makes an angle θ with the line of dead points, is $= \dfrac{p\,A\,L}{2240} \cdot \dfrac{\sin(\theta+\phi)}{\cos\phi}$.

The values of the last term for successive angles of $22\frac{1}{2}°$, when the length of the connecting-rod is twice that of the stroke, which is its usual value in marine engines, are given above.

the connecting-rod is taken into account, may be obtained geometrically as follows (Fig. 406) :—

The twisting moment is $= Q \times OD = \dfrac{P}{\cos\phi} \times OE \cos DOE$

but by similar triangles, $DOE = DGO = \phi$

$\therefore$ twisting moment $= \dfrac{P}{\cos\phi} \times OE \cos\phi = P \times OE.$

Consequently, for any angle of the crank, the ordinate of the curve of twisting moments will be proportional to the part O E of the vertical radius intercepted by a line drawn in the direction of the connecting-rod at the instant. For the corresponding angle of crank, if the connecting-rod were infinite, the twisting moment would be proportional to the vertical dotted line C F.

To obtain the combined twisting moments for the several cylinders of an engine, the curves representing the twisting moments for the respective cylinders are first drawn, the curve for the second cylinder commencing at the point corresponding to the angle its crank makes with the crank of the

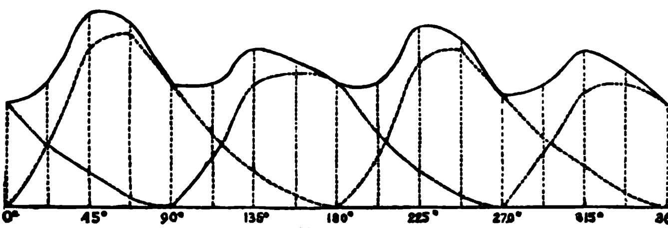


FIG. 407.

first cylinder, and so on. The ordinates of the curve showing the total twisting moment represent the sum of the ordinates of the twisting moments exerted by the several cylinders at the given angles.

An example of this is given in Fig. 407, which shows the twisting moments on the crank-shaft of a two cylinder compound engine with connecting rod four cranks long.

(B.) *EFFECT OF THE INERTIA OF THE RECIPROCATING PARTS OF THE ENGINES.*

In the foregoing examples the pressure of the steam on the piston at any time, as shown by the indicator diagram, has been assumed to represent the pressure on the crank-pin for the corresponding part of the stroke. This, however, will be considerably modified by the inertia of the reciprocating parts of the engines. The angular velocity of the crank-pin is supposed to be uniform, the crank moving through equal angles in equal times, and the length of the connecting-rod infinite. The pistons, rods, &c., however, are at rest at the beginning of each stroke, their velocity gradually increases up to mid-stroke, when it reaches a maximum, after which the velocity decreases and becomes zero again at the end of the stroke.

The acceleration during the first half of the stroke can only be produced by the exercise of a pressure, which pressure must evidently be deducted from the steam pressure on the piston in order to obtain the actual pressure on the crank-pin. During the second half of the stroke, when the motion of the piston, &c., is being retarded by the action of the crank-pin, the work

accumulated during the acceleration is given out in pressure, which has to be added to the steam pressure on the piston in order to get the total pressure on the crank-pin. The pressure producing the retardation is practically equal to that producing the acceleration, the only difference being due to the alteration in the velocity of the piston at opposite ends of the stroke resulting from the obliquity of the connecting-rod.

The effect of the inertia of the reciprocating parts, therefore, is to alter the distribution of the pressures on the crank-pin during the stroke; so that whilst, if friction be neglected, the total force that acts on the crank-pin during the stroke is equal to that acting on the piston, yet in the first half of the stroke the pressures on the crank-pin are less, and in the second half greater, than those on the piston. It is therefore clear that the variation of strains on the crank-shaft will be considerably affected from this cause, and it is important that the weights of the reciprocating parts should always be taken into account in constructing the curves of twisting moments.

From the laws of motion we know that if a body of weight W move from rest under the action of a constant acceleration force R, at the end of t seconds

$$R\,t = \frac{W}{g}\,v$$

where v = velocity in feet per second, and $g = 32{\cdot}2$, the accelerating force of gravity.

If s be the space through which the body has moved in the time t,

$$v = \frac{2s}{t} \;\therefore\; R\,t = \frac{W}{g}\cdot\frac{2s}{t};\ \text{or } R = \frac{W}{g}\cdot\frac{2s}{t^2}$$

In the case of an engine the acceleration for the first few degrees of the crank is practically uniform, but it soon begins to diminish, and at mid-stroke, when the velocity has reached a maximum, and is for the instant uniform, the acceleration becomes zero. As a matter of fact the acceleration is absolutely greatest at the beginning of the stroke, and diminishes gradually up to half-stroke; though for the first two or three degrees the rate of diminution is so slow that for our present purpose it may be regarded as practically constant for the time under consideration. The difference between velocity and acceleration must be borne in mind. The acceleration is a maximum when the velocity is least, and becomes zero when the velocity reaches its maximum, and is for the instant uniform. This may be clearly seen when we remember that it is only *change of velocity* that requires the exertion of a force, so that, neglecting friction, when a body is moving uniformly no force is required to keep it in motion.

The inertia of a body may be defined as the property it has of, when at rest, remaining at rest, or when in motion continuing to move with uniform velocity unless acted on by some external force.

In the case of the reciprocating parts of the machinery of a steam-engine, if we apply the formula

$$R = \frac{W}{g}\cdot\frac{2s}{t^2}$$

we find that for the first degree of revolution, during which the acceleration is practically uniform,

if L = length of crank in feet, and n = number of revolutions *per second*, $s = L \times {\cdot}0001523$; ${\cdot}0001523$ being the versine of an angle of one degree.

Also, $t = \dfrac{1}{360\,n}$; therefore, the force R necessary to produce the given ccceleration in the moving parts

$$= \frac{W}{g}\cdot\frac{2s}{t^2} = \frac{W}{g}\cdot\frac{2L \times {\cdot}0001523}{\frac{1}{(360\,n)^2}} = 1{\cdot}227\ W.L.n^2$$

If this be divided by the area of the piston in square inches, we shall get the equivalent pressure on the piston to produce the acceleration; which pressure must be deducted from that given by the indicator diagram at the beginning, and added to that given by the diagram at the end of the stroke, in order to obtain the actual pressures exerted on the crank-pin.

If p represents this pressure, and A = area of the piston,

$$p = \frac{R}{A} = 1{\cdot}227 \frac{W L n^2}{A} \text{ or } = {\cdot}00034 \frac{W L N^2}{A}$$

In order, therefore, to obtain the pressures on the crank-pin at the respective portions of the stroke, a diagram showing the pressures required to accelerate or retard the reciprocating parts must be combined with the indicator diagram. The amount of acceleration will gradually diminish from the commencement of the stroke and become zero at mid-stroke, when retardation commences, and then gradually increases until at the end of the stroke it becomes equal to the acceleration at the beginning of the stroke. The diagram showing the accelerating and retarding forces will therefore be similar to Fig. 408.

Let A B represent the length of this diagram. A D = B E = p = pressure due to acceleration at the beginning, and retardation at the end, of the stroke.

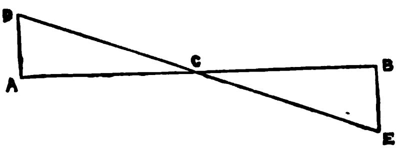


FIG. 408.

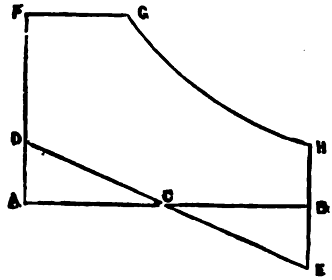


FIG. 409.

The straight line joining D and E will show the variation in the accelerating forces produced by the inertia of the reciprocating parts throughout the stroke. At C, the centre of the stroke, the acceleration is zero.

The combination of the two diagrams may perhaps be best illustrated by its application to a theoretical diagram. Let A F G H B, Fig. 409, be the indicator diagram; the initial effective pressure on the *piston* being A F and the final B H. A D = B E = pressure due to the acceleration of the reciprocating parts, calculated as before explained. Join D E. Then the effective pressure on the *crank-pin* at the beginning of the stroke is D F, and at the end H E. The varying pressures on the crank-pin during the stroke are given by the diagram D F G H E D, and are evidently much more uniform than those given by the indicator diagram A F G H B, which shows the steam pressures on the piston only. At the middle of the stroke the pressures on the piston and on the crank-pin are equal to each other.

In the case of an ordinary indicator diagram, the form of which is much less regular than that of the theoretical diagram, in order to obtain the pressure on the crank-pin, the acceleration diagram should be drawn as directed, and then the several ordinates of the indicator diagram decreased or increased by the values of the accelerating or retarding forces at the respective parts of the stroke, to form the diagram that gives the pressures on the crank-pin.

Fig. 410 is the diagram of twisting moments on the crank-shaft of the engine whose curve is shown in Fig. 407, when allowance is made for the inertia of the reciprocating parts.

In vertical engines the pressures on the crank-pin are also affected by the dead weight of the moving parts. This is equivalent to the addition during

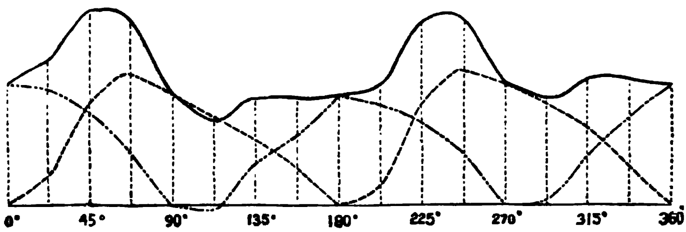


Fig. 410.

the down-stroke, and subtraction during the up-stroke of a pressure equal to the total weight of the parts divided by the area of the piston.

If p_1 = this pressure, $p_1 = \frac{W}{A}$ and the pressures to be subtracted from or added to the pressures on the indicator diagram will be those given by the acceleration diagram, plus or minus $\frac{W}{A}$, according as the stroke is down or up.

In the previous investigation of the forces produced due to the inertia of the moving parts the two following assumptions were made—

(1) that the connecting-rod was of infinite length,

and (2) that the velocity of the crank-pin was uniform.

Of these (2) is practically true, as engines have usually more than two cranks, and the turning moment does not vary much.

As regards (1), however, the small length of the connecting-rod alters the accelerating and retarding forces due to the inertia, the straight line D C E in Fig. 411 now becoming a curve as shown D_1 L E_1. The force is increased at

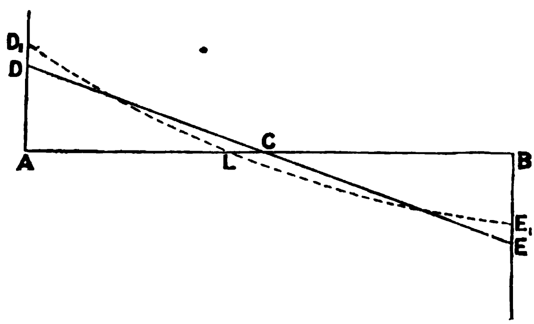


Fig. 411.

the beginning of the stroke and decreased at the end; the point of no force L being moved, for the maximum piston velocity is not when the piston is at the middle of its stroke, but when at a distance C L from it, which is approximately when the connecting-rod and crank are at right angles.

To find the values of the forces due to inertia, we must find the acceleration of the parts, and then the forces being proportional to the accelerations we can find the forces. First, to find the velocities of the piston and crank. The piston motion is not now harmonic, although as before, we suppose the crank to turn uniformly.

Then if I G is perpendicular to G O B and C I is O C produced, I is the instantaneous centre of the connecting-rod G C, and therefore

$$\frac{\text{velocity of G along G B}}{\text{velocity of C along crank circle}} = \frac{\text{I G}}{\text{I C}} = \frac{\text{O E}}{\text{O C}}$$

as the triangles O C E and C I G are similar.

$$\text{Also } \frac{\text{O E}}{\text{O C}} = \frac{\sin \text{O C E}}{\sin \text{O E C}} = \frac{\sin (\phi + \theta)}{\cos \phi} = \tan \phi \cos \theta + \sin \theta$$

and as ϕ is small $\tan \phi = \sin \phi$ approximately

$$\therefore \frac{\text{O E}}{\text{O C}} = \sin \phi \cos \theta + \sin \theta \text{ approximately.}$$

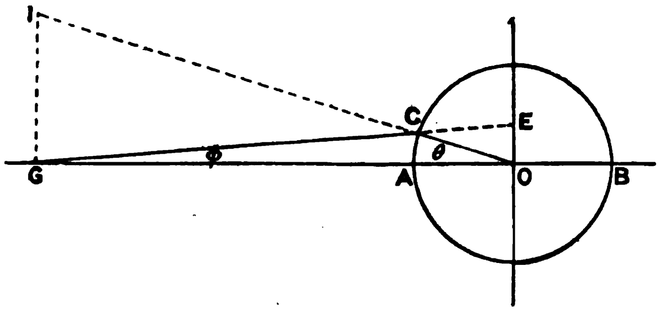


FIG. 412.

Let the connecting-rod $= n r$ where $r =$ radius of crank, and V_o and V be the linear velocities of crank-pin and piston respectively.

$$\text{Then } \frac{\text{G C}}{\text{O C}} = \frac{\sin \theta}{\sin \phi} = \frac{n r}{r} = n, \text{ or } \sin \phi = \frac{\sin \theta}{n}.$$

Substituting these values we have

$$\frac{\text{Vel. of G along G A}}{\text{Vel. of C along crank circle}} = \frac{V}{V_o} = \sin \phi \cos \theta + \sin \theta,$$

$$\text{or } V = V_o\left(\frac{\sin \theta \cos \theta}{n} + \sin \theta\right) = V_o\left(\sin \theta + \frac{\sin 2\theta}{2n}\right)$$

The acceleration is given by differentiating with respect to t, the details of which we omit.

If W = weight of the reciprocating parts, the force due to the acceleration $= \frac{W}{g} \times$ acceleration, from which the accelerating or retarding force

$$= \frac{W}{g} \frac{V_o^2}{r}\left(\cos \theta + \frac{\cos 2\theta}{n}\right)$$

or, if $p =$ the equivalent pressure per square inch of the piston, N the number of revolutions per minute, and A = area of piston in square inches,

$$p = \frac{W V_o^2}{g r A}\left(\cos \theta + \frac{\cos 2\theta}{n}\right)$$

$$= \cdot 00034 \frac{W}{A} r N^2 \left(1 + \frac{1}{n}\right) \text{ at the beginning of the stroke when } \theta = 0^\circ$$

$$= - \cdot 00034 \frac{W}{A} r N^2 \left(1 - \frac{1}{n}\right) \text{ at the end of the stroke when } \theta = 180^\circ.$$

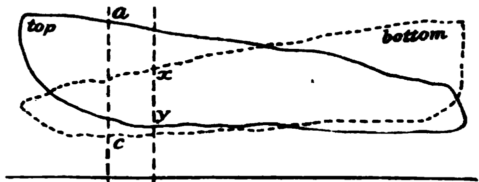
Also the point L can be obtained, for it is the point of no acceleration, and the piston has therefore a maximum velocity, which will occur approximately when the crank is at right angles to the connecting-rod, and therefore

$$C\,L = \sqrt{n^2 r^2 + r^2} - n\,r$$

Knowing the weights of the reciprocating parts of an engine, the curve of inertia can be constructed and applied to an indicator diagram, exactly as in Fig. 409, obtaining A D and B E from the expression just found and filling in the curve through L. The curve of turning moments is then obtained as before. In calculating the weights of the reciprocating parts the weight of the connecting-rod should be considered as concentrated at its two ends, the proportion of weight at each end being inversely as the distance of the end from the centre of gravity of the rod, so that less than one half of the weight is reciprocating.

An example of an actual diagram converted to show the forces acting on the crank and parts of the connecting rod and piston rod is given below. The indicator diagrams being as in Fig. 413, the forces acting due to steam pressure, as at Fig. 315, are transferred to a straight line base, the curve for the down stroke being drawn above, and that for the up stroke below, the base line. The converted diagrams are shown in Fig. 414; thus $a\,c = a'\,c'$ for the down stroke part, and $x\,y = x'\,y'$ for the up stroke part; the same curve of inertia will now serve for both strokes. Draw the inertia curve F F, taking the weights of piston and rod and the proportion of weight of connecting rod as described above; then the ordinates intercepted between this curve and the diagrams $t\,t$ and $b\,b$, for top and bottom respectively, give the pressures to be used on page 484 for drawing the curve of twisting moments. By drawing other curves of inertia on the diagram, the actual forces on parts of the engine, such as the small end of piston rod or gudgeon pin bolts, can be obtained. In each case the correct weight of reciprocating parts must be taken, *e.g.* in the case of the screwed piston rod end the weight taken would be that of the piston with the small length of rod beyond it, and the nut. In the case of the gudgeon pin bolts, the weight of the remainder of piston rod and crosshead must be added.

Fig. 413.



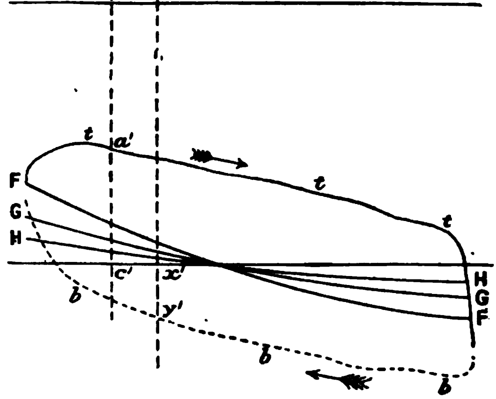


Fig. 414.

Figs. 413 and 414 are from a torpedo-boat destroyer. H H is the base line for the stress on the piston rod, G G that for the gudgeon pin bolts. In these two cases only the lower curve or that for the up stroke is used, these parts having no stress on the down stroke.

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